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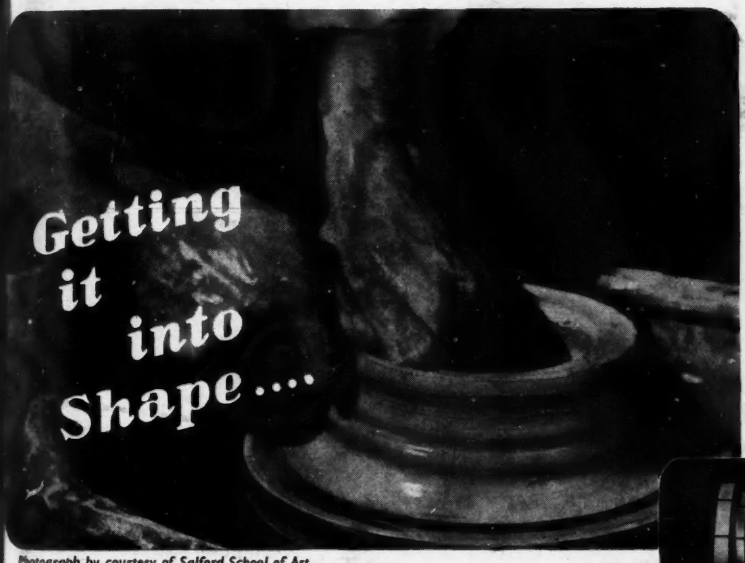

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HIGH PRESSURE VESSELS FOR THE CHEMICAL INDUSTRY

By E. SIEBEL and S. SCHWAIGERER. (From *Die Technik*, Vol. 1, No. 3, September, 1946, pp. 114-118, 7 illustrations.)

MODERN production of synthetic chemicals is largely based on application of high pressure and high temperature processes requiring containers and vessels of highly specialized design. Progress in Germany since 1935 is described in this article.*

(1) *Materials and Service Conditions.* In producing synthetic ammonia from hydrogen and nitrogen the working pressures vary from 100 to 1000 atmospheres and the temperatures from 400 to 600 deg. C. With these and other chemical processes severe attacks of high pressure hydrogen on the steel cylinders are common experience, often made worse by the presence of sulphuric and other gases. Bad effects are already noticeable at 300 deg. C., rapidly increasing with higher temperatures and resulting in reducing the carbon content of the steel, the methan formed under high pressure loosening the steel structure and producing brittleness of the material. These dangers have been counteracted by using alloyed steels which showed good resistance to the action of high pressure hydrogen at high temperatures. Chromium steels with up to 7 per cent chromium content proved successful, with certain additions of molybdenum, tungsten, vanadium or titanium.

(2) *New designs of vessels.* The metallurgical remedies as described above were, however, combined with a completely new approach to the design of these containers. A comparatively thin-walled inner cylinder is made of hydrogen resisting alloy steel while the high pressure is taken care of by outer cylinders of boiler quality material. The forged vessels made in one piece by very heavy hydraulic presses were thus superseded by multi-layer vessels of various types.

(a) The forged thick-walled cylinders were replaced by several thin-walled cylinders fabricated by welding and shrunk on to the high-alloy core. These compound structures are safe against internal high pressure which radially presses the inner against the outer cylinders. Fig. 1 shows the longitudinal section of such a vessel of 700 mm. internal diameter and 5000 mm. length without cover, designed for 325 atmospheres working pressure. The cylindrical wall of 119 mm. total thickness consists of the core cylinder and a number of 6 to 8 mm. thick cylindrical layers, each of them welded, the longitudinal seam machined flush and X-rayed, then heated to 400 deg. C. and shrunk on to the preceding layer. This is a rather involved procedure, which was replaced by the use of a smaller number of thicker-walled cylinders assembled on the core cylinder and longitudinally welded in position. Only the core cylinder can be X-rayed. The shrinkage due to welding will assist in closing the gap between individual layers, but pressure tests have shown that even after partial plastic deformation unequal sharing of the load by these cylinders has to be reckoned with.

(b) This disadvantage is avoided with *Banded Vessels.* They consist of a core cylinder with a series of profiled bands wrapped spirally around the core. This construction is based on the well-known fact that, with a hollow cylinder under internal pressure, the axial stress is only half of the tangential stress. (This is why the longitudinal seams in a cylinder are twice as heavily loaded than the circumferential seams). The profiled bands take the tangential load and, owing to their stepped surface, transmit also part of the axial load, thus relieving the core cylinder which can accordingly be built as a fairly thin shell of high quality alloy steel. Figs. 2 and 3 give one suitable band profile and show the arrangement of the band layers when wrapped (Schierenbeck's design). The bands are electrically heated to 500-600 deg. C., and wrapped round the core in staggered arrangement thus covering the gaps of the preceding layer. Profiled rollers exert pressure during wrapping so that bosses and grooves, assisted by shrinkage when cooling down, form a solid wall. Heavy lathes of suitable length are required for the wrapping process.

It has been shown by many pressure tests that these vessels can be safely loaded to capacity. They show ample deformation before failure. Manufacture, however, must be careful and the wrapping process has to be strictly supervised.

The performance of pressure vessels built from welded cylinders in several layers as described under (a) can be greatly improved by filling the hollow space between individual layers by a medium capable of

*Many additional facts and figures are to be found in a recently published Intelligence Report:

FIAT Final Report No. 577, Office of Military Govt. for Germany (U.S.), Survey of the Leading Manufacturers of Pressure Vessels, by ROY W. CLARK.—Editor, ENGINEERS' DIGEST.

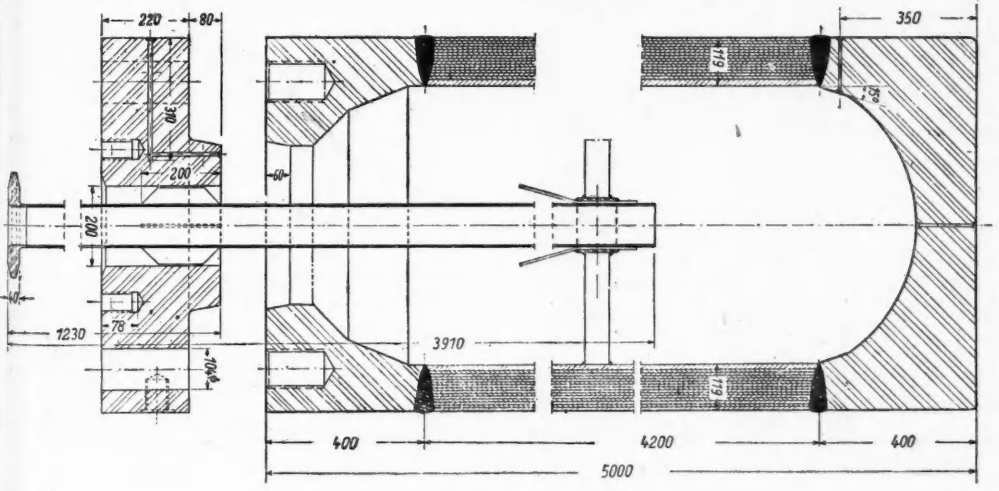


Fig. 1. **Welded multi-layer vessel.**

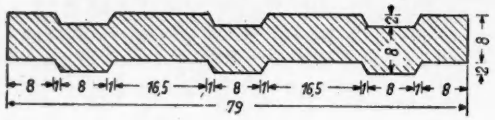


Fig. 2. **Band profile.**

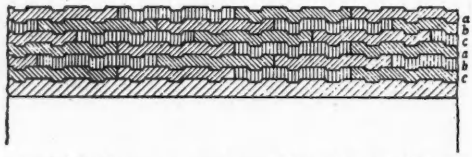
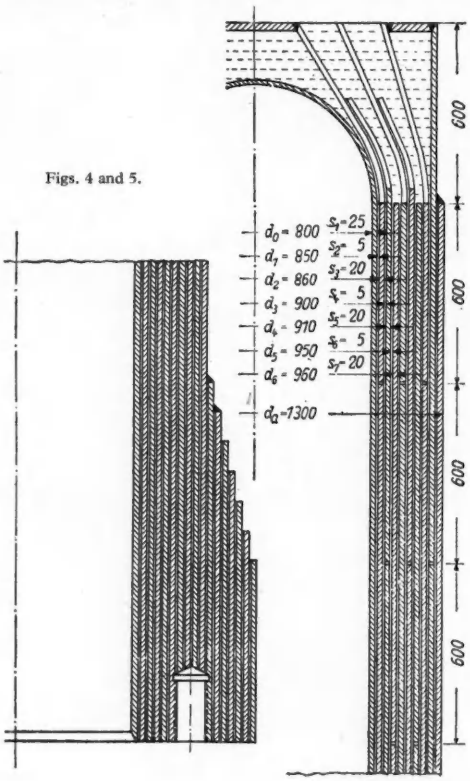


Fig. 3. **Wall of pressure vessel consisting of wrapped band layers.**

transmitting stresses. A sort of liquid cement glue has been injected by pressure. After setting, this compound provides ample bond between the steel cylinders and shear stresses can be transmitted, and there is no longer any need for connecting the shorter length cylinders by circumferential welds. Ample play between concentric cylinders is thus permissible, assembly becomes easier and machining tolerances can be generous. The individual cylinders must be sufficiently long and placed in staggered position. Such an arrangement is illustrated in Figs. 4 and 5. The pressure vessel, of 250 mm. wall thickness, is designed for 700 atmospheres service pressure. Ten cylindrical layers provide cover for the circumferential seams at the third points. Only the core

Figs. 4 and 5.



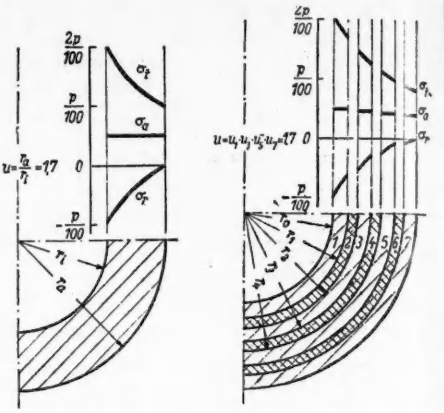


Fig. 6. (**left**) **Stress distribution in solid wall cylinder.**
Fig. 7. (**right**) **Stress distribution in multi-layer vessel.**
σ_t denotes the tangential stress
σ_a ,, the axial stress
σ_r ,, the radial stress.

shell and the outer cylinder have their circular seams welded.

(3) *Basis of calculation.* At any point of the cylindrical shell under internal pressure the resultant stress is made up from a tangential, an axial and a radial component. Its maximum occurs at the inside face :

$$\sigma_t = \frac{1{\cdot}73p}{100} \cdot \frac{u^2}{u^2 - 1} \quad .. \quad .. \quad (1)$$

while the minimum tension at the outside face is :

$$\sigma_a = \frac{1{\cdot}73p}{100} \cdot \frac{1}{u^2 - 1} \quad .. \quad .. \quad (2)$$

For solid wall vessels with a factor of safety of 1·8 against yielding at the inner face at service temperature, the permissible working pressure is :

$$p = \frac{\sigma_F}{1{\cdot}8} \cdot \frac{100}{1{\cdot}73} \cdot \frac{u^2 - 1}{u^2} \quad .. \quad (3)$$

where p . . . working pressure

$$u = \frac{r_a}{r_i} = \frac{\text{outer radius}}{\text{inner radius}}$$

σ_F . . . yield stress.

With multi-layer vessels there can be no guarantee of perfectly tight fitting, and the above equations have therefore to be modified. The inner layers will have to take a bigger share of the load than the outer cylinders. The resultant stress at the inner face will be :

$$\sigma_i = \frac{1{\cdot}73p}{100} \cdot \frac{u^2}{u^2 - 1} \cdot \varphi \quad .. \quad .. \quad (4)$$

where the coefficient φ will be 1·0 to 1·1 with machined cylinders, and 1·1 to 1·2 if they are left unmachined. For the core cylinder the factor of safety against yielding can be reduced to $S_F = 1{\cdot}35$, but the welded seam requires the introduction of another coefficient v to represent its reduced strength. The permissible working pressure is therefore :

$$p = \frac{\sigma_F v}{S_F} \cdot \frac{100}{1{\cdot}73\varphi} \cdot \frac{u^2 - 1}{u^2} \quad .. \quad (5)$$

Conditions are rather different with the banded vessel. Here the core cylinder will be in compression due to the shrinkage of the bands, and the bands will be

pre-stressed in tension. With increasing internal pressure the first signs of yield will therefore occur at the outer bands, while there is no risk of yielding at the inner surface. A factor of safety of 1·6 against plastic yield is recommended.

Figs. 6 and 7 give a comparison of the stress distribution in a solid wall cylinder and in a multi-layer vessel, the ratio of outer to inner radius being 1·7 in both cases.

(4) *Conclusion.* Two new types of high pressure vessels have been developed in Germany since 1935. In contrast to the solid wall vessel, made by forging, pressing or welding, the new types have their cylinders built up in layers. The vessels consisting of concentric layers of welded cylinders cannot be expected to give perfectly tight fit, while the latest type, the banded vessel, ensures "monolithic" behaviour under internal pressure. Quality of workmanship can be checked by stress measurements under water pressure test loads.

RUSSIA

MEASUREMENT OF CURRENT IN ELECTROLYTIC BATHS

By B. A. Novikov. (From "*Promyshlennaya Energetika*," No. 6, 1946, pp. 9-10, 2 illustrations.)

The methods generally used in an electro-deposition to ascertain whether current distribution is satisfactory are very rough-and-ready. Touching the clamps to see if they are heating, observing the bubbles around the cathode (in galvanizing) and so forth. It is difficult with such methods to spot poor electrode contacts, incipient short circuits and other defects, all of which lead to waste of power and production time.

To overcome this difficulty a special measuring instrument has been devised—the "Electroschup" or "electric feeler." It is made in two forms: type No. 1 (Fig. 1) and type No. 2 (Fig. 2). In a bath which is intended to have a total current of 5000 amps. and has 25 anodes, each anode should be carrying about 200 amps. If one of the anodes carries very much more current, there must be a short circuit and the instrument will immediately locate the seat of the trouble. If, on the other hand, the current is below normal, there must be a bad contact either between the bus-bar and the anode or at the cathode clamps. Again, the instrument will quickly indicate not only the existence of an irregularity but also its location.

The "Measuring Electroschup" (No. 1) consists of two soft iron strips (1) joined by a brass plate (5). Higher up a moving-iron system (8) is pivoted between the soft-iron strips with the pivot at (13), a pointer (11) and a restraining spring (12). Above part (5) everything is encased in a metal and glass case (3, 4) and the underside of (5) is rubber lined to cushion and insulates it from the conductor to be tested (16). A socket for a handle is provided at (7).

Current passing through the electrode (or clamp)

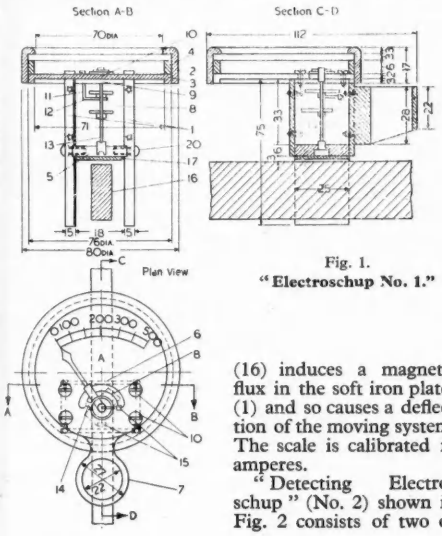


Fig. 1.
"Electroschup No. 1."

(16) induces a magnetic flux in the soft iron plates (1) and so causes a deflection of the moving system. The scale is calibrated in amperes.

"Detecting Electroschup" (No. 2) shown in Fig. 2 consists of two or

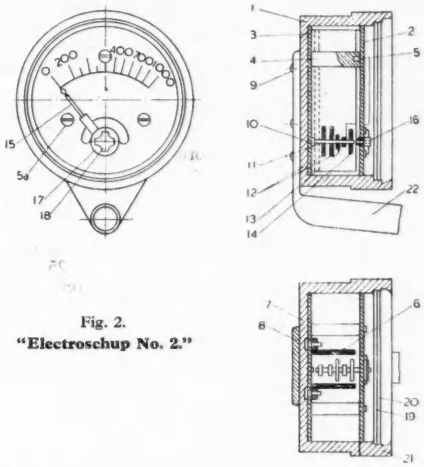


Fig. 2.
"Electroschup No. 2."

three moving-iron armatures (12) on a pivot (11) with a pointer (15) and restraining spring (14) located between brass base-plates (2 and 3). On either side of the moving system is a small angle-iron (6). When the instrument is in use, these angle-irons are parallel to the electrode or clamp and perpendicular to the bus-bars, so that they render the system insensitive to the bus-bar current and also protect the system from the influence of adjoining electrodes.

The angle-irons, it will be noted, extend practically right across the instrument in both the cross-sectional views, thus thoroughly protecting the moving system from the magnetic effects of all currents except those of a conductor which lies parallel to them and is centrally located with respect to them, i.e. of the electrode or holder under investigation, and for this one electrode they act as a magnetic circuit which helps the flux to influence the moving system. Thus it is found that, at a convenient measuring point 6 to 8 ins. from the bus-bar in the example previously quoted, the instrument was practically unaffected by the 5000 amps. in the bar while giving accurate indication of electrode currents 25 times smaller.

The outer casing (1) has a glass top (20) and a tubular handle (22). The base-plates (2 and 3) are held apart by three distance-pieces (4 and 5a). The scale can be adjusted to suit requirements by modifying the shape and position of the armatures.

There are two ways of using this instrument ·

(1) By touching each electrode in turn and measuring the current in it.
(2) By merely passing the instrument over the electrodes in rapid succession, when an excessive or insufficient swing over any individual conductor is easily spotted.

SWITZERLAND

ON THE INSTALLATION OF MECHANICAL STOKERS IN EUROPEAN STEAM LOCOMOTIVES

By J. P. BAUMGARTNER. (From *Bulletin Technique de la Suisse Romande*, Vol. 72, No. 23, November 9, 1946, pp. 306-307.)

INTRODUCTION.

THE dimensions of European steam locomotives are steadily increasing. Many engines built in the past 20 years have fire-grate areas of 4·5 to 5 sq. m. (48-54 sq. ft.). As the amount of coal manually supplied to the grate is limited to the quantity which one man can handle, and as this quantity is practically invariable, it must be realised that there is no point in increasing the dimensions of the fire-box if manual stoking is retained, and that the only way of obtaining a higher engine power beyond a certain limit will be to provide locomotives with mechanical coal stokers. The following questions immediately arise :

(1) What is the limiting fire-grate area which one man can keep supplied and beyond which mechanical stoking is required ?

(2) To what extent will the overall thermodynamic efficiency of the locomotive be affected by the introduction of mechanical stoking ?

LIMIT OF FIRE-GRATE AREA WITH MANUAL STOKING.

For a given locomotive, the rate of hourly combustion A, expressed in weight of coal per unit area of grate per hour, corresponds to a rate of vaporisation V, expressed in weight of steam per unit area of grate per hour, according to the equation :

$$V = A \frac{m}{n} r \qquad .. \qquad .. \qquad (1)$$

where m denotes the calorific value of the coal,
n the total heat of the steam leaving the boiler,
r the thermal efficiency of the boiler.

It is known that r decreases with an increase in the hourly rate of combustion A, according to the linear equation

$$r = a - b\,A \qquad .. \qquad .. \qquad (2)$$

where a and b are constants depending on the boiler characteristics.

Substituting (2) into (1), A results in the value

$$A = \frac{am - \sqrt{a^2m^2 - 4bmnV}}{2bm} \qquad .. \qquad (3)$$

The average values of a and b, derived from numerous tests on European locomotives, are : $a = 0{\cdot}83$, $b = 0{\cdot}0004$.

V amounts to about 3,000 kg. per sq. m. per hour for a locomotive working at full load, corresponding to a medium rate of vaporisation of 60 kg. per sq. m. of total heating surface per hour.

To find A from equation (3), two extreme cases should be considered. In the first—extremely favourable—case, a locomotive is assumed to operate with moderate superheating and to burn excellent coal (m = 8,000 Cal. per kg.), which produces steam of a total heat n = 745 Cal. per kg. Hence we obtain

$$A_1 = 425 \text{ kg./sq. m.h. } (87 \text{ lb./sq. ft. h.}).$$

The second case—extremely unfavourable—envisages a low calorific value m = 6,500 Cal./kg. and a high total heat n = 785 Cal./kg., obtained on a locomotive with a high degree of superheating. Equation (3) gives

$$A_2 = 625 \text{ kg./sq. m.h. } (128 \text{ lb./sq. ft. h.}).$$

In normal service and with full load, the hourly rate of combustion A varies therefore between 425 and 625 kg./sq. m. h.

An average stoker is able to put 1,500 to 1,800 kg. of coal per hour on the fire-grate, i.e. 25 to 30 kg. (55 to 66 lb.) per minute. Higher figures should not be taken into account, as they can be maintained only for very short periods. Hence the values for the surface of the fire-grate adequately served by manual stoking vary between

$$G \text{ max.} = \frac{1{,}800}{425} = 4{\cdot}25 \text{ sq. m. } (45{\cdot}7 \text{ sq. ft.})$$

$$\text{and } G \text{ min.} = \frac{1{,}500}{625} = 2{\cdot}40 \text{ sq. m. } (25{\cdot}8 \text{ sq. ft.})$$

If the grate area appreciably exceeds the value of 4·25 sq. m., it is impossible, in ordinary service and with manual stoking, to obtain the expected locomotive power. It follows, too, that there must exist quite a number of European locomotives never able to produce, in normal service, the power for which they were designed.

This leads to the conclusion that *it is necessary to provide every locomotive with a mechanical stoker, if its fire-grate reaches, or exceeds, an area of 5 sq. m.* (54 sq. ft.).

Moreover, it appears advisable to study and to experiment with the application of mechanical stoking in the cases of grate areas of 3·5 to 4·5 sq. m., because there cannot be any doubt that the use of a mechanical stoker is justified in a great many of these cases.

INFLUENCE OF MECHANICAL STOKING ON THE THERMODYNAMIC EFFICIENCY OF A LOCOMOTIVE.

The mechanical stoker most widely used is the American designed " Standard." A worm takes the coal from a bunker to a distribution table placed in front of the door at the interior of the fire-grate. Steam jets guided in various directions move it from there on to the grate itself and ensure a layer of even thickness. The worm is driven by a small steam engine which is completely enclosed and splash lubricated.

The consumption of the mechanical stoker amounts to 2·3 per cent of the water vaporised by the boiler at low combustion rate (A = 250 kg./sq. m. h.) and to 1·4 per cent at very high rates (A = 1,100 kg./sq. m. h.), the average being 2 per cent.

The mechanical stoker does not permit the use of rough coal of all sizes. The size of the fuel should not exceed 75 mm. The stoker breaks the coal and serves the grate with pieces of 5 to 25 mm. in diameter. The draft easily carries away the small grain and, in order to avoid greater losses of unburnt coal, fuel with a high content of hydrocarbons (35 to 40 per cent, against 25 to 30 per cent with ordinary rich coal) should be used which gives the coal particles time to be burnt on the grate before being blown off by the draft. To obviate losses, it might even be necessary to provide a rather long brick vault or a special kind of grate. Grate bars of the " Hudson " type are most frequently met with. They reduce the sectional area for the passage of the combustion air to 16 to 20 per cent of the grate area, as compared with 35 to 50 per cent in the case of the grate bars generally used with manual stoking. They permit a low fire to be maintained, the grate being covered by only a few centimetres of fuel, but against this they have the disadvantage of increasing the resistance to the passage of air through the grate ; in consequence, a

greater vacuum is needed in the smoke-box in order to maintain a certain rate of combustion. With the same type of exhaust it is necessary, therefore, to reduce the blast-pipe section by one-tenth to one-fifth, if the locomotive is equipped with a mechanical stoker and an appropriate fire-grate. For a given rate of combustion, the back-pressure at the exhaust which is inversely proportional to the cross-sectional area of the blast-pipe will rise from 20 to 55 per cent. This reduces the indicated power of the locomotive 1 to 5 per cent and proportionally increases its specific consumption of steam and fuel.

The application of a mechanical stoker and of a suitable grate affects the thermal efficiency of any given boiler. It slightly reduces it at lower rates of combustion, but increases it at very high loads.

Considering all particularities of the mechanical stoker, i.e.

(1) The consumption of its steam engine ;

(2) The necessity to throttle the blast-pipe ;

(3) The influence on the thermal efficiency of the boiler, it can be stated that its application to a modern European locomotive increases the fuel consumption per indicated horse power-hour about 5 per cent in the case of usual combustion (250 to 500 kg./sq. m. h.) and by about 3 per cent at a combustion rate of 750 kg./sq. m. h. and that this increase tends to disappear at very high loads. At the same time, the total thermodynamic efficiency developed at the draw-bar of the locomotive is reduced 0·6 per cent

From the economical point of view, this small reduction is more than compensated for by the saving due to fuel of lower quality burnt in the locomotives provided with mechanical stokers.

HOLLAND

THE NOTCHED BAR IMPACT TEST ACCORDING TO SCHNADT

By J. A. HARINGX. (From *De Ingenieur*, Vol. 58, No. 50, December 13, 1946, pp. Mk. 15-17, 8 illustrations.)

As a result of the bending effect occurring in the notched bar impact test of the usual type, there exists in the smallest section a tension stress side by side with a zone of compression stress. Quite a good part of the amount of work to be performed by the bob of the impact tester must be expended upon the deformation of material situated in the compression zone. This deformation will not be present in the case of brittle fracture, and it must be doubted if the so-called impact test values in such a bar can at all be taken as a reliable measure of the tendency of the material investigated towards brittle fracture.

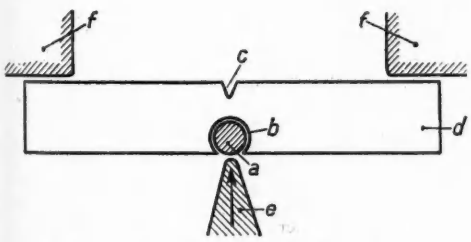


Fig. 1. **Impact test according to Schnadt.**
(a) Hardened steel pin, (b) hole in test bar, (c) notch, (d) test bar, (e) knife edge of bob, (f) supports.

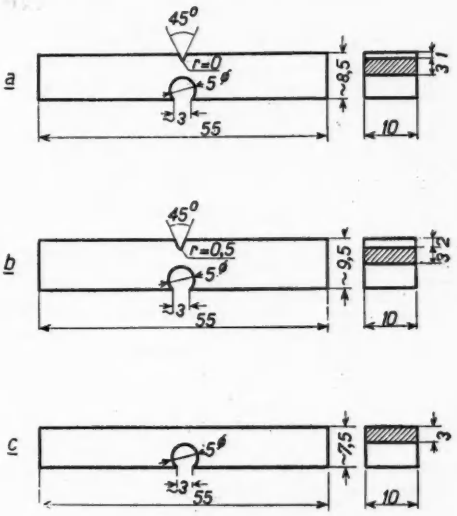


Fig. 2. **Test specimens according to Schnadt.**
(a) $r = 0$, (b) $r = 0{\cdot}5$, (c) $r = \infty$

For this reason Schnadt has resorted to removing the material in the compression zone by drilling a hole and inserting into it a hardened steel pin " a " as shown in Fig. 1. During bending the centre line of this pin, which fits loosely into the partly open hole " b " of the test bar " d," will more or less play the role of a pivot ; so that the material of the test bar is exclusively subjected to tensile stress. In addition to the fact that the pendulum bob now strikes against the hardened steel pin instead of against the softer material itself, a considerable advantage accrues from the fact that the test bars are definitely broken, even if no notch is present at all. This makes it possible to use quite different bottom radii for the notch with an identical fracture area. In practice Schnadt uses only three different bottom radii, namely $r = 0$, $r = 0{\cdot}5$ mm. and $r = \infty$ (Fig. 2), the respective specific work-of-fracture values being K_0, K_1, and K_i. It may be added that Schnadt suggests to produce the notch with $r = 0$ by indenting the bar in the press with the use of a hardened knife edge ($r \sim 0{\cdot}02$ mm.). With the exception of cast iron and zinc practically all materials show high K_i values.

The smaller the bottom radius r, the more closely the state of all-sided compression is approximated, and the greater the tendency to brittle fracture will be. Whilst under normal conditions a high K_1 value at the respective test temperature will be a guarantee for a satisfactory toughness of the material in question, in the case of high stress concentration a high K_0 will be required. But if no stress concentrations are to be expected, then the existence of a high K_1 value alone will be sufficient.

RUSSIA

ELECTRIC ENERGY BY HIGH-FREQUENCY CURRENTS

By G. I. BABAT. (From *Journal of Technical Physics*, Vol. 16, No. 5, 1946, pp. 555-564, 9 illustrations.)

ONE of the basic problems of heavy-current applications of high-frequency, such as H.F. traction, is the low-loss transmission over a distance l, considerably exceeding the wave length λ of the electromagnetic wave used.

In a non-contact H.F. traction system the line runs below the track (e.g. led through insulated channels underneath the road surface), and the road cars are fitted with receiving coils inductively coupled with the non-contact traction line. By changing the tuning of the receiving coil the counter-e.m.f. induced in the traction line is altered and thus also the H.F. power received by the vehicle. The maximum power the car can take from the line is proportional to $I_c^2 \omega M$. It is difficult to make the mutual inductance M between coil and line large, while keeping the line current low, in order to reduce the cost of copper and the copper losses, hence the necessity of using high frequencies which, in the author's experience, should be not less than some 10,000 c/s.

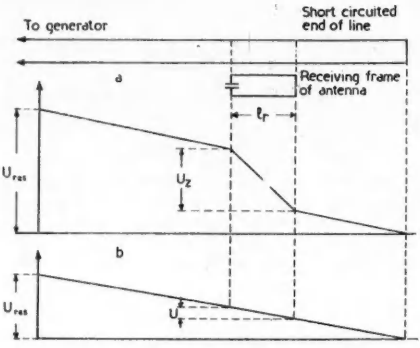


Fig. 1. **Curves of the effective value of the voltage between conductors of a section of a non-contact traction system coupled with the receiving circuit of an H.F. operated vehicle.** (a) Tuned, loaded receiving circuit, (b) short-circuited (non-tuned, no-load condition of receiving circuit).

In the case of a simple two conductor traction system, shown in Fig. 1, the current intensity will be uniform over the whole length when frequency and length of section are such that $\frac{\lambda}{4} \gg l$.

With lengthening of the line, or with an increase in frequency, the uniformity of current distribution will be disturbed, and stationary waves of current and voltage may be set up in the system. Near the nodes of the current wave the transmission of energy by induction from the line to the receiving circuit becomes impossible. Yet it is possible to obtain uniformity of current intensity and potential difference between the two conductors all over the length of the line, for $l > \lambda$, independent of the loading, by the use of parallel and series compensation.

Fig. 2 schematically represents a fully compensated line. The condensers, series-connected into the conductors, compensate the inductive voltage drop, and the self-inductances in parallel with the conductors compensate the capacity current between the conductors of the line. The shorter the distance l_{vC} between the points of connection of the single condensers, and the distance l_{vL} between consecutive inductors as compared with the wave length λ, the better is the relative uniformity of the current in the line.

Alternatively, compensation by distributed capacity is possible, i.e. a line design in which one, or both con-

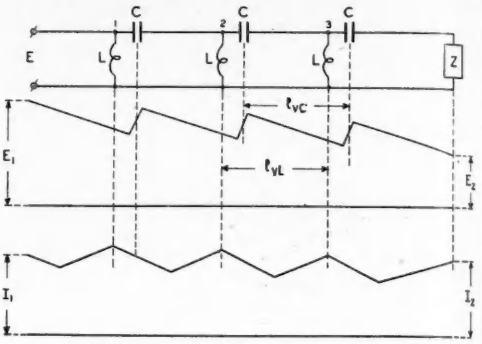


Fig. 2. **Diagram of compensated single-phase traction system with current and voltage curves.**

ductors are subdivided into single sections, insulated from one another and so distributed that their mutual capacitance suffices for compensation as well as for the inductive voltage drop (cf. arrangement of Fig. 3).

The whole system, of rather unusual appearance from the point of view of conventional traction engineering, is split up into short sections insulated from one another, (single vibrators or dipoles). What the given example clearly shows is that the energy is transmitted from the generator to the consumer through the dielectric, and that the conductors merely serve as "guide rails" to direct the flow of energy. The system discussed thus forms an intermediate stage between the conventional L.F. transmission line, and wave-guides or directional aerials.

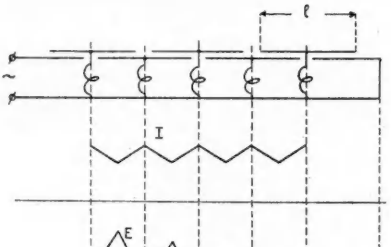


Fig. 3. **Circuit diagram with current and voltage curves of line compensated by distributed capacity, double covering of vibrators.**

Still better current uniformity in the compensated H.F. line is to be achieved by using not only a double covering (see Fig. 3a) of vibrators, but even multiple coverings. Thus, Fig. 4 shows a triple covering of vibrators, wherein each successive vibrator is displaced by a third of its length with respect to the preceding one.

When the capacity between the individual vibrator and all the others of the same polarity does not several times exceed the capacity between the said vibrator and the conductor of the opposite sign, then the current distribution in every individual vibrator will approach the sinoidal, and the length of the vibrators will approximate $\lambda/4$. The total line current is obtained as the sum of the individual vibrator currents (cf. curves in the

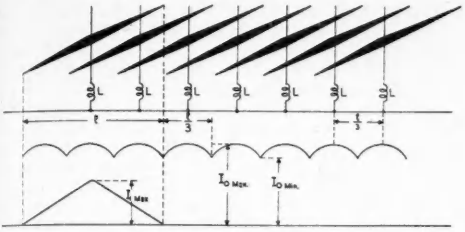


Fig. 4. **H.F. Line of linear vibrators with triple covering (Step equal to one-third of the vibrator length *l*). Below, the curves of current in the individual vibrators and of total line current.**

lower part of Fig. 4). Apart from the number of coverings, the uniformity of the line current depends on the ratio C_1/C_2 being made as large as possible.

Fig. 5 shows a line with four-fold covering. In such systems it is easy to obtain the relation

$$\frac{I_{0\ max} - I_{0\ min}}{I_{0\ max} + I_{0\ min}} < 0{\cdot}05$$

To ensure equality of the current in all the vibrators they must all be arranged similarly.

In Fig. 4 the arrangement is sloping, but where necessary, more complex arrangements are possible (cf. Fig. 3). The vibrators may be distributed in either single or double, vertical or horizontal rows. As the current intensity in the vibrator falls from the centre towards the ends, the vibrators may be made tapering towards the ends for the sake of copper economy, as seen in Figs. 4 and 5.

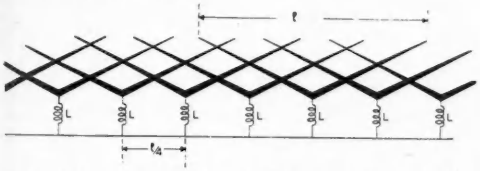


Fig. 5. **Zig-zag arrangement of vibrators with quadruple covering.**

The resonant wave length of a compensated system and the relations between voltages and currents may be found by the methods used in wireless for the calculation of unsymmetrical aerials (aerials consisting of two dissimilar parts). Fig. 6 shows the lay-out of a compensated system. Let C_1 be the capacity per unit length between an individual vibrator and all the adjacent ones, and C_2 the capacity per unit length between the same vibrator and the conductor carrying current of the opposite sign. Fig. 7 then gives the relation between the resonant wave length and the ratio C_1/C_2.

When $C_1/C_2 > 5$, it suffices for all practical purposes to treat the line consisting of single vibrators as a circuit with lumped constants. The capacity of such a circuit is equal to $C_1 \frac{l_v}{2}$ and its inductive reactance, $L_1 \frac{l_v}{2}$. In this case

$$\lambda = 6{\cdot}3\ l_v \sqrt{\frac{C_1}{C_2}}$$

In an uncompensated line having a characteristic impedance Z_l, the maximum voltage between conductors, U_{max}, is related to the maximum line current, I_{max}, by $U_{max} = Z_l\, I_{max}$.

In compensated systems the potential difference

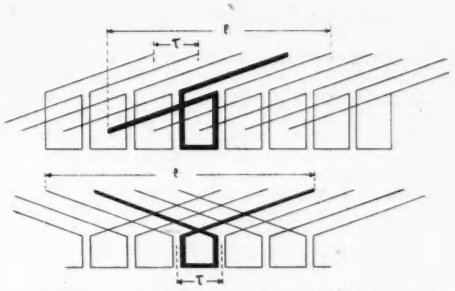


Fig. 6. **Formation of single element of a composite multi-vibrator H.F. line, for the determination of the resonant wave length.**

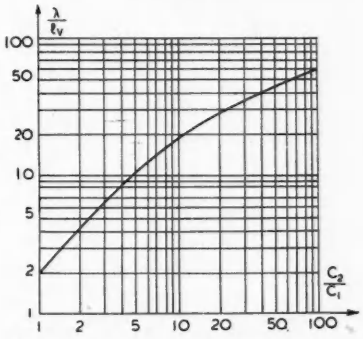


Fig. 7. **Ratio of resonant wave length to length of the single vibrator *l*, in a non-contact traction system, plotted against the ratio of the capacity per unit length between single vibrator and the rest of the conductors of similar polarity to the capacity between vibrator and the conductor of the opposite sign.**

between their single elements (i.e. the individual vibrators is C_1/C_2 times smaller.

Fig. 8 shows a fully compensated three-phase non-contact traction system, whose lay-out is of the zig-zag type, and which produces a travelling field. Such systems ensure full uniformity of the energetization of large areas. On principle, the surface of the permanent track may be larger than λ^2.

It should be noted that the electromagnetic energy emitted by radiation is not determined by the ratio S/λ^2, but by p^2/λ^2, where p is the "polar stress" of the energized surface, i.e. half the distance between the nearest conductors carrying equally directed currents. It follows that even when energizing big areas this may be done with small radiation losses.

Fig. 8 also shows the compensating inductances connected in delta, although they might as well be connected in star, as the total wattless power (kilovars) of the inductances does not depend on the chosen system

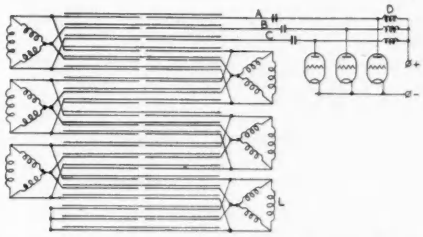


Fig. 8. **Diagram of a compensated H.F. three-phase line**

of connection, the latter determining only the working voltage.

To reduce the losses in a non-contact traction system, it is expedient to divide it into separate sections which are connected to the feeding line only if and when the H.F.-driven cars, to which the energy has to be supplied, are in those sections. Under no-load conditions the sections carry only a small relaying current (principle of the " sleeping line ").

It thus becomes possible to realize any desired relation between length of the section, frequency of the current, electromagnetic field intensity, and feeding voltage by applying the principle of the line subdivided into single vibrators.

Fig. 9 shows different designs of the section, in the particular form of a coil of 4 turns. To compensate the inductive reactance, a condenser may be connected accross the terminals of the section (cf. top diagram, dotted lines). Alternatively, compensation by distributed capacity between the conductors could be used. In this case, the current would show an approximately linear decrease from the centre of every conductor towards its ends (as symbolized in Fig. 9 by the decreasing thickness of the line). The greater the number of vibrators making up the section, the shorter the resonant wave length. In the construction of coils for wireless circuits it is generally desirable to keep the inter-turn capacity as low as possible, but in traction systems this capacity can be used with advantage. In order to obtain the maximum capacity it is expedient to keep those windings as close together as possible between which there exists the greatest potential difference.

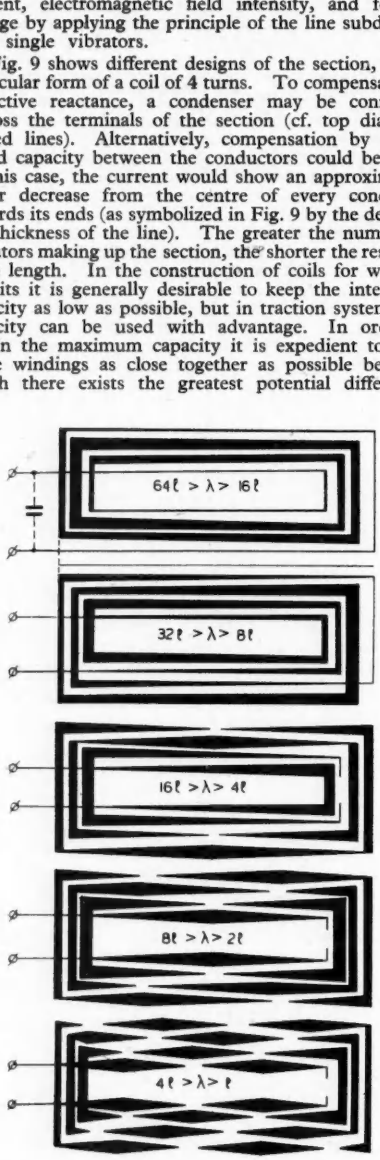


Fig. 9. Alternative lay-outs of the section of a non-contact traction system for different ratios of wave length to length of the section.

Multi-turn traction systems like those shown on Fig. 9 enjoy the advantage of not having an external electrical field giving rise to additional losses in the surrounding medium. In such systems there always exists the useful magnetic field only, i.e. that transmitting the energy to the moving system.

Single-turn traction systems need special screening conductors placed across the system conductors for cutting out the losses due to the electric field component.

Compensation by distributed capacity may also be applied to multi-core cables. Again, copper may be economized by a tapering of the vibrators forming the cable cores.

It is necessary to equalize the voltages in the various vibrator sections of a multi-vibrator line in their connection points to source of supply, and load, respectively. This may be done by means of auxiliary inductances and capacities connected in series with the vibrators.

In fully compensated lines it is important to know the magnitude of the reactive power of the compensating condensers and choke coils relative to the power component transmitted by the line.

In cases where the load impedance Z_a of the compensated line is equal to its characteristic impedance, $Z_l = \sqrt{L/C}$ the total reactive power P_{rL} of the choke coils connected in parallel with the line is equal to the total reactive power of the condensers in series with the line conductors. Thus, the reactive power per unit length of the line, P_r, is equal to the useful power transmitted, P_a, divided by $\lambda/4$. If, however, the load impedance is not equal to the characteristic impedance of the line, then,

$$P_{rL} = \frac{4P_a}{\lambda}\,\frac{Z_a}{Z_l}\;;\; P_{rC} = \frac{4P_a}{\lambda}\,\frac{Z_l}{Z_a}$$

For a load impedance considerably greater than the characteristic impedance of the line, $Z_a \gg Z_l$, we get

$$P_{rC} \ll P_{rL}$$

Such relations obtain for non-contact traction lines underneath the road surface, operated at comparatively low voltages. In this case it will often be possible to work without parallel chokes, and series condensers will be sufficient.

It will be of interest to compare, from the economical point of view, the compensated multi-vibrator line with distributed capacity, with a line with mica condensers in series.

The capacitive reactive power obtainable from a line built up of single vibrators is directly proportional to the square of the potential difference between the vibrators, inversely proportional to the wave length, and also depends on the connection of the system and on C_1/C_2.

Admitting a maximum voltage between line-conductors of 20 kV, it is possible to obtain a capacitive reactive power $P_{rC} = 10^4$ kVA per km. length of the line at a wave length of $\lambda = 5000$ m.

Mica condensers are about 10 roubles per kVA. It follows that the cost of the batteries (of condensers) necessary for the said reactive power would be about 100,000 roubles.

In the given example the use of compensation with distribution constants is economically important, as the splitting up of the line into single vibrators increases the cost of the line by less than 100,000 roubles/km. At higher frequencies the effect is even more marked. Conversely, when reducing the frequency and the working voltage of the underground traction line, the capacitive reactive power per unit length, obtainable from a system with distributed constants, falls.

At voltages lower than 5 kV and wave lengths greater than 5 km. the use of special condensers with solid dielectric is preferable to distributed capacities.

A NEW CRITERION FOR THE DESIGN OF WIRE ROPES

By M. TEN BOSCH. (From *Schweizerische Bauzeitung*, Vol. 128, No. 19, November 9, 1946, pp. 237-239, 3 illustrations.)

AN attempt is made in this paper to arrive at reliable methods of calculating the life of wire ropes, and suggestions are made for an amendment in existing by-laws governing the use of such ropes.

The design of ropes is based on the most unfavourable conditions of loading. These vary greatly and depend on the function which the rope is intended to perform, e.g. running or stationary, and external atmospheric influences such as corrosion and icing. Normal design requirements are that danger of complete failure, i.e. fracture, is to be avoided with a certain margin of safety, but in certain cases lower limits of loading are called for, such as in overhead lines for transmission of electric power, which must not be stressed beyond the elastic limit.

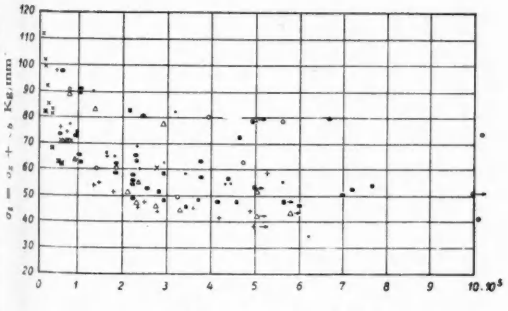
Generally speaking steel ropes are subject to higher intensities of loading than other parts of machines from the moment of manufacture and throughout their useful lives. In addition to tensile and bending stresses throughout the rope, very high compressive stresses occur at points of contact between wires, often combined with a sliding movement. Frequent fractures which occur near points of support of hauling, carrying and traction ropes, and in parts of ropes which never run over a pulley, indicate that the cause of failure may be due to vibrations set up in starting, to braking or possibly to wind. Very high tensile stresses are thereby induced, causing in turn dangerous compressive stresses between wires and supports, and between adjacent wires. Wedged-shaped grooves of driving pulleys often add considerably to these compressive stresses. A further cause of high stress is the waviness of profile of a stranded rope which leads to a sudden transfer of the supported load from one strand to the next, i.e. impact forces of high frequency.

Although the many factors leading to overstressing usually prevent a diagnosis of the final cause of any particular failure, the following causes of probable failure in the various ropes are enumerated :—

Standard crane ropes, wires of equal thickness, with twists in opposite directions or twists in one direction only ; compressive stresses increased by wedge action of sliding supports, and by alternating bending stresses.

Parallel-strand ropes : alternating bending stresses.

Long-hauling ropes : oscillations cause first wire failures. See Herbst [1] who reports on failures halfway between supports.



n_3 = number of stress reversals to failure.

Fig. 1. **There is no law of relationship between maximum stress uder load ($\sigma_t = \sigma_z + \sigma_b$) and number of stress reversals to failure (n_B). [6].**

Note. The different symbols for showing the points plotted in Fig. 1 each correspond to a series of tests carried out by Scoble [4] and Woernle [5].

Reference is made to standard practice [2] in which ropes are designed only with reference to maximum tensile loads, although the knowledge of the static tensile stress which would cause failure (K_z) does not enable the designer to estimate the useful life of a rope.

Neither does the additional calculation of bending stresses ($\sigma_b = \delta E/D$) (see Fig. 1).

Compressive stresses at points of contact can be calculated by using Hertz's equations [3] if the design of the rope (angle of twist) is known. Even if allowance is made for the effect of these stresses, in addition to tension, bending and sliding, it does not seem possible to calculate therefrom the useful life of a rope.

For twenty years Scoble [4] and Woernle [5] have made systematic tests on wire ropes to investigate the various factors which lead to rope failures, but they did not arrive at reliable methods of design.

To make a systematic use of experimental results recourse is often made to the principle of similarity, which is based on the fact that physical phenomena are independent of the system of mensuration and can be expressed by means of equations whose terms are dimensionless. The useful life of ropes is such a phenomenon and must be capable of description by means of such dimensionless numbers.

There are, in this case, two categories of these dimensionless numbers. The first defines the mechanism and the second, the use of the mechanism. The first category contains definitions :

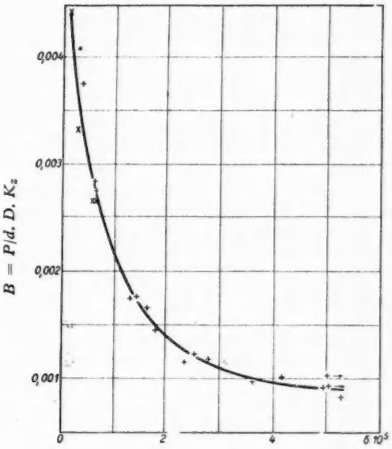
(a) of the design of the rope (e.g. single, double, treble twist—applying to strand, group of strands, rope respectively) ;

(b) of the type of twist (in opposite directions, in same direction, parallel, or Tru-lay) ;

(c) of the design of each strand (No. of layers—2, 3 or 4) (equal wire thicknesses, Seale, or Warrington, etc.) ;

(d) of the conditions of contact between the rope and the pulley ;

(e) of the conditions of lubrication ;



n_B = number of stress reversals to failure.

Fig. 2. **The useful life of crane ropes with twists in opposite directions (6 × 37 wires) is clearly defined by the dimensionless number B.**

etc., all of which are in themselves dimensionless numbers and do not contribute to the formation of new dimensionless terms.

The second category contains terms relating to :

(a) the tensile force in the rope (P) ;
(b) the diameter of the rope (d) ;
(c) the diameter of the wire (δ) ;
(d) the diameter of the pulley (D) ;
(e) the material (defined by the ultimate tensile stress (K_z), the Brinell hardness number ((H_B) etc.) ; all of which are grouped together to form one or more dimensionless terms.

Drucker and Tachau [6] who have taken the first step in this direction have arbitrarily formed a number B by grouping together P, d, D and K_z thus :—

$$B = \frac{P}{d.D.K_z} \qquad .. \quad .. \quad (1)$$

Although the tensile strength of the rope (K_z) in itself is no criterion, the number B containing it has been proved to define the useful life of ropes provided all factors enumerated above as falling into the first category remain unaltered. (See Fig. 2.)

Scoble and Woernle have chiefly tested standard ropes which contain wires of equal thickness. Standard specifications [7] comprise the following three standard types (twists either in opposite directions or in same direction) :

Type-A rope, $6 \times (1 + 6 + 12) = 6 \times 19 = 114$ wires of diameters from ·4 to 1·0 mm.

Type-B rope, $6 \times (1 + 6 + 12 + 18) = 6 \times 37 = 222$ wires of dias. from ·4 to 1·3 mm.

Type-C rope, $6 \times (1 + 6 + 12 + 18 + 24) = 6 \times 61 = 366$ wires of dias. from ·7 to 1·5 mm.

Fig. 2 refers to Type-B ropes with twists in opposite directions, the experiments having been confined to $\delta = ·7$ to 1 mm., and $d = 16$ to 22 mm.

Fig. 2 shows that the number of stress reversals (a measure of the useful life),

$$n_B = \text{function } (B) \qquad .. \quad .. \quad (2)$$

Experimental results have been plotted logarithmically in Fig. 3, and the following simple equation is deduced therefrom :—

$$n_B = 6·2/B^2 \qquad .. \quad .. \quad (3)$$

For certain given criteria falling into the first category above, and for a given tensile stress in the rope, the useful life depends therefore on the product of $d.D$ and not, as assumed in the by-laws, on the ratio d/D.

The diameter of the wire, δ, does not appear in the definition of B. As, however, the ratio of $\delta/d = 1/22$ is constant in the case of standard B-Type ropes [7] B can be expressed as

$$B = \frac{P}{22\ \delta\ D\ K_z} \qquad .. \quad .. \quad (1a)$$

δ and d, in this case, have the same influence on the useful life, and δ is considered preferable because it introduces a relationship to the minimum bending stresses.

Substituting from equations for bending stress ($\sigma_b = \delta\ E/D$) and tensile stress ($\sigma_z = P/\frac{\pi}{4}\delta^2\ i$), B becomes

$$B = \frac{\sigma_z\ \sigma_b}{K_z\ E}\ \frac{i\ \pi/4}{22} \qquad .. \quad .. \quad (1b)$$

or if an apparent factor of safety $S = K_z/\sigma_z$ is introduced

$$B = \frac{i\ \pi/4}{22}\ \frac{\delta}{D}\ \frac{1}{S} \qquad .. \quad .. \quad (1c)$$

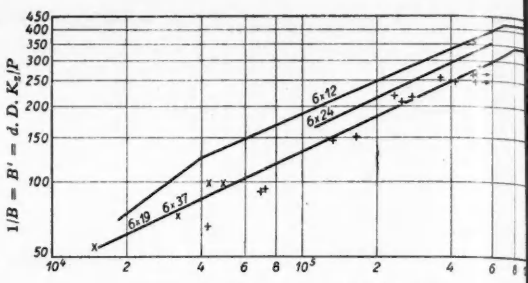


Fig. 3. The dimensionless number B' also defines the useful life of other types of ropes with twists in opposite directions.

Equations (1), (1a), (1b) and (1c) and any conclusions derived therefrom apply in the first instance to Type-B ropes with twists in opposite directions, but if the factor $i\ \pi/4$ is extended to include further numbers describing the design of the rope (the number of layers, a, and the number of strands, b) the following table is obtained :—

	Type-A	Type-B	Type-C
No. of wires, i ..	6 × 19	6 × 37	6 × 61
d/δ	16	22	28
No. of layers, a ..	2	3	4
$\frac{i}{b}\ \frac{\delta}{da}$	·6	·55	·55 (const.)

It appears from this table that the design of the rope has little influence on its useful life, a fact confirmed by experiment, as far as standard ropes are concerned. Experimental results on Type-A ropes ($d = 8$ to 16 mm.) agree closely with those on Type-B ropes. No experiments on Type-C ropes have been reported.

From observations it appears that ropes with twists in the same direction have a longer life than those twisted in opposite directions, other factors being equal. There is, however, a greater deviation of plotted observations from the curve drawn through them, and the author suggests that ropes twisted in the same direction may be more sensitive to small alterations in the angle of twist. The reciprocal of B (symbol B') is conveniently adopted because B' increases with n_B.

$$n_B = 6·2\ (B')^2 \qquad .. \quad .. \quad (3a)$$

If an attempt is made to eliminate the ultimate tensile strength K_z from the definition of B or B', recourse may be taken to the constant ratio, confirmed empirically, of K_z to H_B the Brinell hardness number. For carbon steels

$$K_z = ·36\ H_B \qquad .. \quad .. \quad (4)$$

and a new number can be introduced

$$B_1 = B'/·36 = d\ D\ H_B/P \qquad .. \quad (5)$$

Hence $$n_B = 6·2\ (·36\ B_1)^2 = ·8\ B_1^2 \qquad .. \quad (6)$$

The advantage of the use of B in research is that an investigation of the one number B will give a complete picture of an extensive range of the four factors P, d, D and K_z which would otherwise require separate investigations.

The author gives a numerical example of the design of a workshop crane to carry 10 tons on four ropes. The first step is to deduce from economical and other considerations a required useful life and calculate therefrom n_B, the number of stress reversals in each rope carrying 2·5 tons. By means of equation (1b) the product $\sigma_z\ .\ \sigma_b$ is calculated from n_B (for the particular conditions assumed in the example $\sigma_z\ .\ \sigma_b = 1320$ kg.^{2}/mm.^{4}). Ten standard ropes are tabulated, σ_z for each is calculated dividing 2·5 tons by the cross-sectional area of metal in

the rope, σ_b for each is then obtained by dividing 1320 by σ_z, and by substituting σ_b and δ in the equation $\sigma_b = \delta E/D$, the diameter of the pulley appropriate for each rope is finally determined. Whereas the by-laws would only permit one rope to be used, the present method allows any one of the series of standard ropes to be used, each with an appropriate pulley diameter (which decreases with increasing rope diameter).

It is emphasized that equation (3) only applies to similar ropes used under similar conditions. (In this example the ropes are A-, B-, or C-types, and used as running ropes subjected chiefly to tension and bending). Fig. 3 shows, however, that equation (3), with different coefficients, can be applied equally to other similarly constructed ropes.

New equations should be derived for ropes of entirely different design (e.g. Seale, Warrington), or different uses of ropes.

Symbols.

δ = diameter of wire
d = diameter of rope
D = diameter of pulley
i = number of wires
a = number of layers of wires in strand
b = number of strands in rope
σ_t = total stress
σ_z = tensile stress
B,B' and B_1 = dimensionless numbers.
σ_b = bending stress
K_z = ultimate tensile stress in rope
H_B = Brinell hardness number
P = actual tension in rope
n_B = number of stress reversals in bending (affording a measure of the useful life)
S = apparent factor of safety

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SWITZERLAND

ADVANCES IN CRYSTAL FILTER CONSTRUCTION

By H. Biefer, H. Keller and B. Matthias.

(From *Brown Boveri Mitteilungen*, Vol. 33, No. 8, August, 1946, pp. 214-218, 12 illustrations.)

INTRODUCTION.

The use of crystals at frequencies both higher and lower than formerly possible is described; and the development of artificial crystals permitting the design of crystal filters with wide pass-bands is outlined.

CRYSTALS.

Various crystals suitable for artificial production, were investigated[1], in particular potassium phosphate and ammonium phosphate. Their only disadvantage, relative to quartz, lay in the temperature coefficient of the resonant frequency; for it is not possible (as it is for quartz) to obtain, by suitable choice of cut, a zero coefficient for a particular temperature. It can, in fact, be shown mathematically that this is theoretically impossible for their type of crystal structure.

If the use of thermostats is to be avoided, two possibilities remain:

(a) To alter the temperature coefficients by changing the atomic structure of the crystals, or
(b) To seek new crystals, whose structure is such as to give, at least in theory, the possibility of obtaining zero temperature coefficient (by choice of a particular direction for the cut.)

Both were examined, with the following results:

(a) Potassium and ammonium phosphate act, at room temperature, like normal piezo-electric crystals, their resonant frequency rising with falling temperature, due to hardening of their material.

With still further reduction in temperature, however, there are departures from this rule. Near the boiling point of liquid air, potassium phosphate has dielectric and piezo-electric characteristics similar to those of Rochelle Salt [2]. Polarization increases with falling temperature and, at the Curie-point, spontaneous polarization appears. Polarization opposes a further increase in hardness [3], so that, at a certain temperature above the Curie-point, the hardness and also the resonant frequency are at a maximum. This maximum (on account of its horizontal tangent) is identical with a zero temperature coefficient. The temperature of this maximum for potassium phosphate lies at about —70 deg. C., while for ammonium phosphate it is near —55 deg. C. In other words, they both are useless in practice.

Empirically and theoretically it appears that the temperature of this maximum increases with increase of radius of the cation. The NH_4 group of the ammonium phosphate must be replaced by stereochemically similar ions of greater radius. Thallium and rubidium might replace the NH_4 group but this would lead to thallium and rubidium phosphate, which are known to crystallize in a system of lower symmetry than the tetragonal and which is not piezo-electric. It was found possible, however, by appropriate treatment in the growth of the crystals to produce a previously unknown modification—a rubidium phosphate isomorphic with potassium and ammonium phosphate, having similarities to Rochelle Salt and a Curie-point about 30 deg. C. above that of potassium phosphate. The temperature at which the

temperature coefficient of the resonant frequency passes through zero is raised by about the same amount.

Thallium phosphate crystallized in the monoclinic-prismatic form, but mixed crystals of ammonium and thallium were obtained containing up to 33 per cent Mol thallium. For these, too, the maximum moved to higher temperatures with increasing thallium content. It was thus possible to reduce the temperature coefficient for (NH_4, Tl) mixed crystals, at room temperature by 75 per cent, without special orientation of the crystal cut.

(b) Of newly discovered crystals, a series of isomorphic, strongly piezo-electric crystals of compounds of glucose, which crystallize well in the quartz crystal system, is mentioned.

Unlike potassium and ammonium phosphate, the temperature coefficient of the resonant frequency varies markedly with the direction of cut; while for any direction at room temperature it is usually smaller than can ever be obtained with crystals of the tetragonal system. These experiments must certainly lead to cuts possessing negligible temperature coefficient at normal temperatures.

These crystals are important because they permit the design of filters with band widths which are relatively broad even in the audio-frequency range. Thus ammonium phosphate has a range of resonance approximately eight times that of quartz.

MODES OF VIBRATION AND RANGE OF FREQUENCY.

For potassium and ammonium phosphate the piezo-electric effect is greatest in the direction of the principal axis and the shear deformation is greatest in a plane perpendicular to that (Fig. 1). In a section cut obliquely

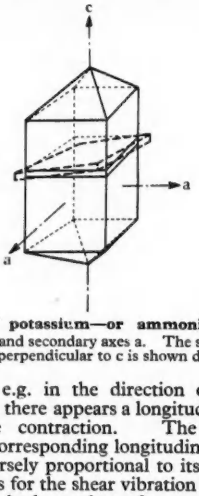

Fig. 1. **Crystal of potassium—or ammonium-phosphate.** With principal axis C and secondary axes a. The shear deformation of a plane perpendicular to c is shown dotted.

from this plane, e.g. in the direction of a diagonal instead of a shear, there appears a longitudinal dilatation and a transverse contraction. The fundamental frequency of the corresponding longitudinal vibration of the section is inversely proportional to its length, and a similar result holds for the shear vibration of the section. With ordinary methods, a plate of one centimetre side gives fundamental frequencies of the order of 100 kc/s. The frequency range is limited in one direction by technical difficulties in the production of large crystal plates, and in the other because damping due to the crystal mounting increases notably as the crystal size decreases. To overcome this, crystals are excited to flexural vibrations for lower frequencies and to vibrations in their thickness for higher frequencies. To produce flexural vibrations, a bar is cut perpendicular to the principal axis and making acute angles with the secondary axes, and provided with two pairs of electrodes, connected crosswise in parallel. Fig. 2 shows that the oppositely directed electric fields acting on the two halves of the bar produce a dilatation in the one half, a contraction in the other half of the bar, and so set up a bending moment. With such oscillators, frequencies down to 20 kc/s are attainable. For still lower

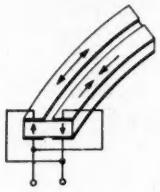

Fig. 2. **Piezo-electric bending.** Electric fields are shown by single arrows, deformations by double arrows.

Fig. 3. **Crystal for specially low frequencies.**

frequencies the form shown in Fig. 3 is used. Here the narrow section performs flexural vibrations and the massive ends perform torsional vibrations. With such crystals the audio-frequency region of the frequency spectrum, which is important for filter technique, can be reached.

To produce higher frequencies, crystal plates are used which can be excited to vibrate lengthwise in the direction of the plate's thickness or to perform shear vibrations in one of the planes containing the thickness of the plate. This, in the case of potassium and ammonium phosphate, requires an oblique orientation relative to all axes, if the exciting field is, as usual, also

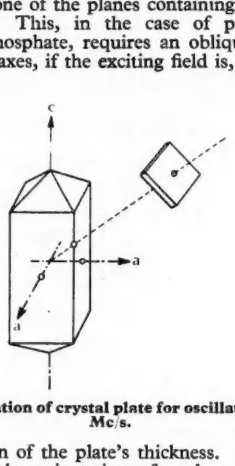

Fig. 4. **Orientation of crystal plate for oscillations at several Mc/s.**

in the direction of the plate's thickness. Fig. 4 shows schematically the orientation of such a crystal. For potassium and ammonium phosphate an electric field in the direction of one axis is associated with a shear in the plane at right angles to it. A field oblique to all axes produces, with its three components in the directions of the three axes, shears in all three normal planes, which in turn have shear and extension components in the direction of the thickness. In this way obliquely oriented plates can be caused to vibrate at frequencies inversely proportional to their thickness. The frequency range is thus extended to several Mc/s.

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SOME CONSIDERATION IN DESIGNING PARTS FOR POWDER METALLURGY

By J. DONAHUE. (From *Mechanical Engineering*, Vol. 68, No. 11, November, 1946, pp. 949-952, 15 illustrations.)

POWDER metallurgy is the technique by which metals or alloys are produced in the finely divided state or powder form, subsequently being formed into finished products by welding together the individual particles of metal powder. Interlocking of the irregularly shaped powder-metal particles is accomplished by pressing in dies that have the shape of the part to be produced. Forming the part to its required shape and density is accomplished by using mechanical or hydraulic presses. It is then heated in an atmosphere-controlled furnace at temperatures below the melting point of one or more of the constituents. Later, if necessary, the part is coined or sized and then heat-treated to a required hardness.

With the end of the war, it quickly became known through the literature how vital the process of powder metallurgy had been to the war effort by saving time and material in the manufacture of many necessary products. The powder-metallurgy process eliminates waste and expensive machining to an extent that would be impossible by any other method. With the knowledge of the existence and value of such a process, many engineers and designers are asking themselves if their concerns have parts that would be adaptable to powder metallurgy. In many cases the answer would be "yes," requiring only minor changes in design.

LIMITATIONS AND ADVANTAGES OF PROCESS.

Powder metallurgy has some limitations. Paramount among these is strength which is generally less in parts produced by this process than when the same parts are produced by the cast or forging method. Largely because of porosity and not as a result of a difference in microhardness, hardness figures seem low in comparison with denser materials. There is a definite limitation to the size and shape of some parts made, owing to the requirement of large presses to obtain the desired compacting pressures, thus resulting in expensive tool and press costs.

The high purity of the metal content of the finished part is one of the advantages of the powder-metallurgy process, thus reducing or eliminating the impurities that are found in metals made by the conventional methods. Purity is controlled by the mixing of selected powders and can be further enhanced at the time of sintering by the removal of oxides through the reducing nature of the atmospheres applied.

The composition of the part produced can be accurately controlled through the use of pure metal powders with little or no change in the composition during the pressing and sintering. Parts containing alloy compositions can be consistently reproduced. Refractory metals such as sintered carbides, tantalum, and tungsten, not possible to fabricate by the conventional methods due to their hardness and brittleness, can be made by the powder-metallurgy process. A wide range of physical properties can be obtained for a particular application, such as density, porosity, and grain size. In small magnets, structures developed may have better mechanical properties than the same material in cast form. The limitation of waste is associated with parts manufactured by powder-metallurgy as there is little or no scrap.

Dimensional tolerances on small and medium-sized parts can be held to $\pm 0{\cdot}001$ in., which is as accurate as parts can be made by the conventional machine operations, such as drilling, milling, planing, and splining keyways. Powdered-metal parts can be laminated with more intricate binding than would be possible by the conventional method of attaching one metal to another. A good example is the combining of plastic materials and metal powders, as is done in magnets. Machined precision parts can be supplanted with powder-metallurgy parts with greater economy and accuracy, thus relieving a shortage of skilled mechanics or machines for parts that require expensive equipment. With the present delay in obtaining this new equipment, powder metallurgy is really the logical answer.

One of the fields of design for powder metallurgy includes structural and functional parts, in which the designer is interested in the shapes obtained. Many intricate shapes requiring costly secondary operations, if produced by the conventional methods, can be produced rapidly and satisfactorily by powder metallurgy at a lower cost, because they adhere to certain fundamental rules of design. On the other hand, much simpler shapes cannot be produced economically by powder metallurgy because they depart from these fundamental rules. Inasmuch as the pressing of the part to obtain the desired density is largely determined by the design of the part and the designer's understanding of the limitations of the pressing process, he can design parts accordingly and take full advantage of what powder metallurgy has to offer.

PRESSING OPERATIONS.

Parts made by the powder-metallurgy process are pressed in a die that has the desired shape. The punches that apply the pressure to the powder in the die operate in a straight line with an upward and downward motion. The lower punch moves downward to the fill position, then rises to a predetermined height, raising the powder to the compression point. At the same time, the upper punch moves downward into the die to its predetermined depth, where both punches compress the powder to the desired density and shape. The lower punch then remains stationary while the upper punch moves upward and out of the die to a clearance height, followed by the lower punch rising to the top of the die, ejecting the part to the table level where it is removed by a sweep, which at the same time fills the die with powder and withdraws. The punches again start moving to complete another cycle. With these operations in mind, the designer must design the tools so that the finished part will have its required shape and density and be ejected from the die without rupturing.

Concentricity must be considered carefully and reasonable limits provided. This is necessary because the part in most cases is formed with tools that have radial fits within each other, and clearance must be provided so that they will not bind or score. If the

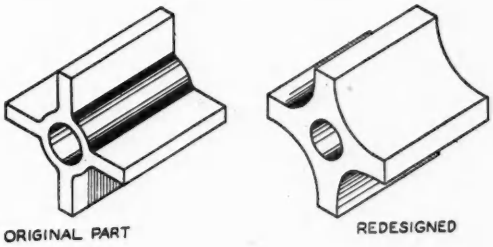

Fig. 1. **Original part and redesigned part for powder metallurgy.**

clearance is too great, a build-up of powder along the die wall will result, causing eccentricity as high as 0·003 to 0·005 in. This may be avoided by not having a larger clearance than is necessary.

The length tolerance should be somewhat liberal in comparison with that necessary in conventional methods of finish, if extra cost of machining the part is to be avoided. As the length of most parts made by powder metallurgy is governed by the fill, done either by hand or mechanically, and regulated by the depth of fill and compressibility of the powder, each part to be designed must be considered individually with the length tolerances on short pieces from $\pm$0·002 to $\pm$0·005 in., while longer pieces will require $\pm$0·010 in. or more.

When parts are designed with radial projections and recessions, as shown in Fig. 1, great care must be taken to avoid deep and narrow splines which make it difficult to obtain complete powder fill, because the powder at the time of filling may bridge across the narrow splines, resulting in a pressed part, with less density and size than is required. The projections will also be weak and porous in structure, while tools made to produce parts with this poor design will rupture when pressure is applied. When parts are designed with these narrow projections, difficulties arise in ejecting the part from the die owing to the increased surface area.

DESIGN CONSIDERATIONS.

Any abrupt change in cross section on the area in the axial direction is not so much the difficulty in pressing to obtain tolerances, as is the expansion or contraction of the part during sintering. This is due to the thin and thick wall sections which will not expand or contract uniformly as would more uniform designs. This change in cross section will cause the part, where the hole is round and the outside not a true diameter, to change in size after sintering, making it difficult to meet the customer's specifications when the tolerances are close.

Bushings. When parts such as bushings are made, the wall thickness, that is, the distance between the outside diameter and the hole, is perhaps one of the designer's most important considerations. It is therefore necessary that a relation between length and diameter be considered. If parts with considerable length and thin walls are pressed, they may not have the desired density due to the powder not reaching the required die fill, i.e. the powder bridging across the narrow opening between the die wall and core rod. The punches will also be extremely weak due to the slenderness of their walls.

Thin Parts. When flat and thin parts requiring high density are pressed, great care must be taken to make sure that the section is not too thin, otherwise the punches may rupture when pressure is applied because of the lack of cushioning effect of the powder. Although sections as thin as 0·015 in. and 0·020 in. have been pressed, in most cases these sizes are obtained by applying a pressure that will give a medium density, sintered, and then coined to the desired density. It is recommended that flat sections of not less than 0·032 in. be pressed.

Shaped Holes. Parts that have shaped holes, keyways, or taper holes, can be made by the powder-metallurgy process by making use of a core rod. When the requirement is for parts having shaped holes, e.g. square, hexagon, half-round, or almost any other shape, the core rod can be shaped to the required design with the corresponding shape in the inside of both the upper and lower punches, so that when the powder is pressed and the part ejected from the die it will have the desired form. When keyways are required, a key is inserted into the core rod with corresponding key splines in both the upper and lower punches, thus helping to shape the keyway when the part is pressed.

Tapered Holes. The forming of tapered holes is more difficult and care should be taken at the time of pressing to prevent the top punch from contacting the taper on the core rod. A short section at the top of the taper on the core rod should have a straight section, so that at the time of pressing the top punch will stop its downward movement on this straight section or above the taper part of the core rod. This will create a flat section in the part which should not be less than 0·032 in., otherwise the core rod may become damaged. This is shown quite plainly in Fig. 2.

Flanged Parts. When making parts that have flanges with a body up to 1 in. diam., the flange diameter should not be more than 1½ times the outside diameter of the part. If parts require flanges greater than 1½ times the outside diameter, a more complicated die-and-punch arrangement must be used to help eject the part from the die, due to the overhang from the main body of the part and the friction on the die wall, which may cause the flange to crack or break. The outside diameter of the part should also have a taper of at least 0·008 in. in each inch of thickness of flange in order to help eject the part from the die. The radius on the inside corner where the flange unites with the body should be at least 0·010 in. This radius will help to strengthen the part and help to eliminate cracking or breaking of the flange at the time of pressing, as shown in Fig. 3.

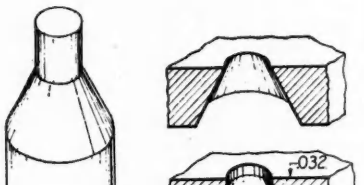

Fig. 2. **Taper design to prevent core-rod damage.**

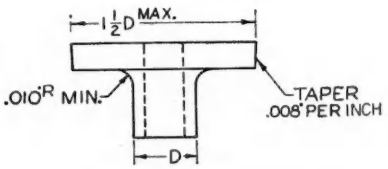

Fig. 3. **Correct-flanged part design.**

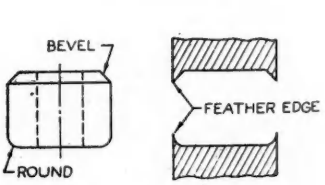

Fig. 4. **Punch considerations.**

Bevels. Parts should not have too large a bevel or radius on their edges, because the bevel or radius on the punch will have a feather edge which may have a tendency to produce a burr at the time of pressing. Also the life of the punches will be short unless great care is taken in the proper choice of tool steels and the subsequent heat-treatment to the correct hardness and toughness, so that they will withstand the pressures at the time of pressing without the feather edge breaking off or turning over. This is shown in Fig. 4.

Parts Requiring Curvature. Parts that require a radius which must be formed during pressing should have a small land on the radius as illustrated in Fig. 5, where the radius on the part begins and the punch stops, in order to avoid a feather edge on the punch. When tools have feather edges, the edge has a tendency to chip or break off because of insufficient material to back it up. When pressure is applied the tool chips or turns over, producing a ragged section on the part.

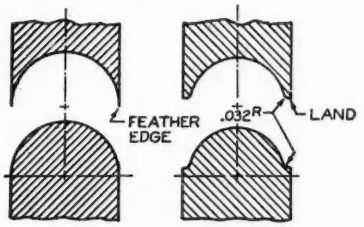

Fig. 5. **Parts pressed with a radius.**

Inserts. Inserts are normally put into the part at the time of pressing but may be pressed into the part after sintering. When a powder-metallurgy part is to be used as an insert, as in plastic moulding, vertical serrations parallel to the axis of pressing can be put on the outside diameter serving the purpose of knurling which otherwise would require a secondary operation. Fig. 6 shows one type of serration.

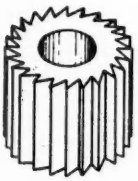

Fig. 6. **One form of serration.**

MACHINING OPERATIONS NECESSARY FOR SOME PARTS.

Parts With Re-Entrant Angles. Parts that have re-entrant angles or curves in the axial direction cannot be pressed because presses now used for powder metallurgy have only a straight up-and-down motion with no side motion or pressure available to compress the powder to make a re-entrant angle or curve. Under these circumstances a secondary operation is required.

Flanged Projections. Parts that have flanges projecting from the centre of the body or more than one quarter of the length from the end, are difficult to form unless complicated dies and punches are used to move the powder to the desired position in the die in order to obtain the required density and help eject the part from the die. Even with complicated dies and punches, it is difficult to say that a perfect part could be produced without breakage or cracking and uneven density. With this type of design a secondary operation would be required, as shown in Fig. 7.

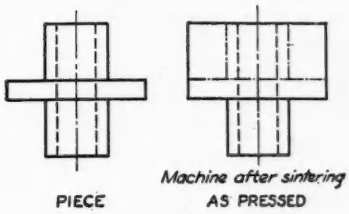

Fig. 7. **Projecting flanges.**

Axial Variations. When parts have axial variations greater than one half of their length, care must be taken to obtain the proper ratio between the die fill and the compressibility of the powder in order to obtain the required density and size.

Parts that have odd shapes and heights greater than one quarter of their length should be redesigned to meet the powder-metallurgy technique, so that the part when finished will meet the required specifications and can be ejected from the die. This is shown in Fig. 8.

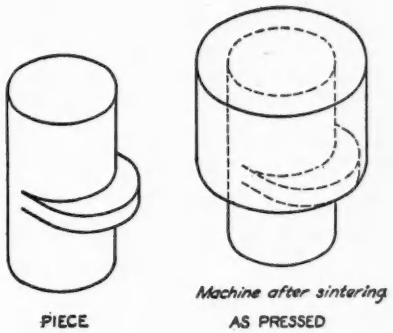

Fig. 8. **Part with projection.**

Depressions, Bosses, and Counterbores. Depressions, bosses, and counterbores should be less than one-quarter axial variation, with the sides tapered to allow the part to be drawn loose from the die or punch. It is good practice not to have depressions or counterbores too deep, because when the powder is pressed to obtain the depressions and counterbores, greater density is obtained than required. This will strain the punches at the time of pressing and if the tool is light in design it will perhaps rupture. With bosses, the opposite reaction takes place; if the required boss is too high and no means of moving the powder to obtain the required density is undertaken, a part with less density is obtained resulting in a weak structure. The designer will have to study each individual part. A good rule to follow is to have the depressions, bosses, or counterbores not over one quarter of the total length or section of thickness with a taper of about 0·008 in. diam. for each inch in depth. Fig. 9 illustrates this.

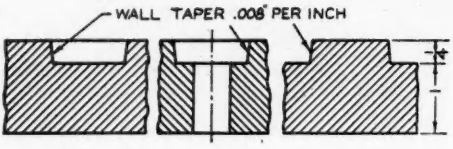

Fig. 9. **Depression, counterbore, and boss.**

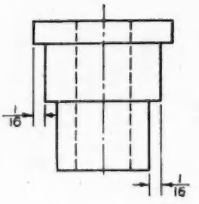

Fig. 10. **Design of stepped part.**

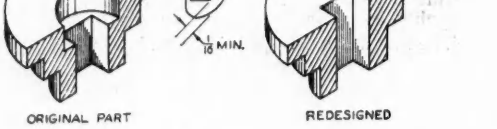

Fig. 11. **Part with square section and counterbore.**
(In the case of this design, the tool to produce the counterbore with a square section would have a weak portion which would rupture during the pressing operation.)

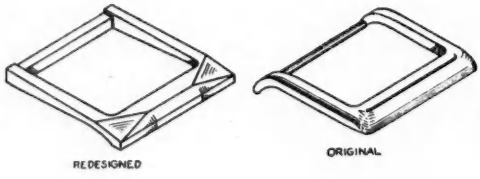

Fig. 12. **Original and redesigned part.**
(To produce a part to meet the specifications, it was necessary to redesign along straight lines, eliminating the curved section. This ensures better pressing and the required density and strength necessary for the part.)

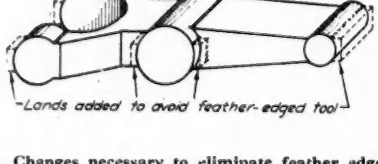

Fig. 13. **Changes necessary to eliminate feather edges on top and bottom punches.**
(The projections or lands on the radius help to eliminate breaking of the edges on the punches during the pressing operation.)

Steps. Where steps are formed in the design, each step should be at least $\frac{1}{8}$ in. larger in diameter than the preceding step. As these steps are sometimes formed either by steps in the die or by a secondary die or punch, enough shoulder should be formed so that there will be sufficient strength in the part, especially when it is necessary to make the step with a secondary die or punch. If the diameter of the step is too small, inadequate powder fill will result, thus defeating the purpose of the design, Fig. 10.

TABLE I. RECOMMENDED TOLERANCES.

Tolerances on length	
±0·005 in.	up to 1 in. long
±0·010 in.	up to 1½ in. long
±0·015 in.	up to 2 in. long
±0·020 in.	up to 2½ in. long
±0·025 in.	up to 3 in. long
Parts pressed and not coined to size-diameter tolerance	
±0·0015 in.	up to 1 in. diam.
±0·002 in.	up to 1½ in. diam.
±0·003 in.	up to 2 in. diam.
±0·004 in.	up to 2½ in. diam.
Tolerances on flange diameters	
±0·004 in.	up to 1 in. diam.
±0·006 in.	up to 1½ in. diam.
±0·008 in.	up to 2 in. diam.
±0·010 in.	up to 2½ in. diam.
±0·014 in.	up to 3 in. diam.
±0·016 in.	up to 4 in. diam.
Tolerances on flange thickness	
± 0 004 in.	up to ½ in.
±0·006 in.	up to ¾ in.
±0·008 in.	up to 1 in.
Concentricity	
0·003 in. total indicator reading up to 1 in. diam.	
0·004 in. total indicator reading up to 1½ in. diam.	
0·005 in. total indicator reading up to 2 in. diam.	
0·006 in. total indicator reading up to 2½ in. diam.	

TABLE II. RATIOS BETWEEN WALL THICKNESS IN RELATION TO DIAMETER AND LENGTH.

Minimum wall thickness, in.	Maximum over-all length, in.	Maximum outside diam., in.
0·032	1	1
0 040	1	1
0·045	1	1
0·055	1	1
0·060	1	1
0 065	1	1
0·070	1	1
0·075	1	1
0·080	1	1
0·085	1	2
0·090	1	2
0·095	2	2

SECONDARY OPERATIONS.

Inasmuch as parts made by the powder-metallurgy technique will not meet all of the requirements of a finished part, secondary operations are sometimes necessary, such as drilling, tapping of holes on the side of the part, tapping of holes formed by the pressing operation, recesses on the end where it would be impossible to press the shape, and the turning of steps that have a diameter only slightly larger than its preceding step.

Along with these factors, the designer must make certain that the design is such as would normally require a considerable cost of machining, as compared to the cost of the part pressed from powdered metals. Reasonable quantities, as well as other factors, are largely related to the design of the part. The more complicated the machining of the part is by conventional methods, the greater the annual saving will be in increased production where parts are made by the powder-metallurgy process.

Tables I and II and Figs. 11, 12, and 13, may assist the designer to determine the proper tolerances and changes in design when designing parts for powder metallurgy.

OUTLINE of SCIENTIFIC RESEARCH WORK on MOTOR CAR DESIGN

By E. A. TSUDAKOV. (From *Automobilnaya Promyshlennost*, 1946, No. 1, pp. 14-19 and No. 2, pp. 8-11.)

I. ENGINES.

THE author submits the outlines of a short term program of research. With respect to selection of types, acceptance of the best foreign designs is suggested, together with the development of own designs to comply with national peculiarities of production and utilization, and, in particular, in accordance with the general road transport requirements of the U.S.S.R. In view of the absence of type selection as it exists in capitalist countries the "perspective selection of types" is extremely important. The following stages in its development are proposed: (a) analysis and classification of conditions under which motor vehicles are used in the different fields; (b) development of a system of grading to establish a relation between the design of a motor vehicle and its performance in use; (c) establishment of basic motor vehicle types (according to load, dynamics, road or cross-country performance and other operating properties); (d) analysis of the trend of development of the design of the motor vehicle and of its individual components (at home and abroad); (e) development of standards for individual types of motor vehicles—with the maximum possible standardisation of components.

First of all it will be necessary to concentrate on heavy lorries, tractors with trailers, and quite generally to supply lorries with trailers, and also to produce large 'buses. This will greatly reduce the price for equal load carrying capacity and the amount of material required, and also reduce the number of drivers.

Research work of special urgency should be carried out on the improvement of the design of motor vehicle engines, i.e. improving economy, reliability, life and specific power output of engines. These problems cannot be solved without simultaneous research work on fuels and oils. The economy of the motor vehicle engine can be raised by: (a) improving the economy of the petrol engine; (b) introducing Diesel engines; (c) using fuels which are cheaper and more plentiful—gas, solid fuel and liquid fuels not made from petroleum.

IMPROVING PETROL ENGINE ECONOMY.

One well established method is the increase in the compression ratio. This involves attention to many other problems. The change-over to aluminium alloy cylinder heads will be essential and experimental engines of this type ought to be built immediately. A most suitable alloy appears to be Silumin as it does not require any finishing operation. In general, the compression ratio of Russian automobile engines is lower than that of the foremost foreign engines so that there are great possibilities in this direction.

Another method is in improving the fuel supply system. This can be achieved by: (a) improving mixture distribution; (b) insuring optimum preheating of the mixture for the different operating conditions; (c) ensuring optimum atomisation of the petrol at all operating conditions; (d) ensuring economical (lean) mixture at all operating conditions except full throttle.

A suitable shape of the intake manifold not only has a beneficial effect on economy but also increases engine power. Extensive research work should be carried out on this subject, and also on down-draught carburettors and multi-carburettor engines.

The problems of preheating must be examined in conjunction with the fuel requirements. The preheating of the vehicle engine before starting in cold weather is of great importance.

Atomisation depends on the design of the carburettor. Modern carburettors fail in this respect at part throttle. Work should be carried out on the development of carburettors with improved atomisation, with two-fold control and especially on variable venturi carburettors.

With respect to operation on lean mixture, experiments have shown that improving economiser design may result in an additional petrol economy of 3-10 per cent. To obtain this, a carburettor would have to be designed in which the economiser is cut out only after the throttle has been fully opened. Furthermore, shutting off the idling jet at part throttle operation yields 2-4 per cent petrol economy, apart from reducing oil dilution when coasting, especially with the clutch in. It is essential that carburettors and intake manifolds are tested in engines and vehicles under variable operating conditions.

Using an automatic gearbox would permit the engine to be operated continuously at its highest efficiency. Petrol injection directly into the cylinders may greatly improve economy, and may also enable lower octane fuels to be used.

IMPROVING IGNITION.

Work on this problem will have to be directed towards improving ignition timing and towards increasing spark intensity. Experiments have shown that improved spark timing may yield 3-4 per cent economy. Perhaps it would be best to relate automatic timing to the first appearance of incipient knock.

Increasing the intensity of ignition would be of still greater value. The greater the volume of fuel-air mixture initially ignited, the leaner may be the mixture through which the flame can be relied upon to propagate. The use of pre-chamber ignition seems especially promising. With this arrangement, a small volume at a chemically correct mixture strength ensures a very active ignition of the bulk of the lean mixture. Other possibilities are: injecting an easily inflammable fuel into the compressed lean mixture, high frequency ignition, etc.

ENGINE WITH POWER RESERVE.

Possibilities in this direction are based on the fact that, generally, motor vehicle engines are rarely required to develop their full power. Theoretical examinations show that a suitably designed engine with power reserve ought to give up to 30-40 per cent fuel economy.

To achieve this aim, several ways are open: (a) accumulating excess engine power, to be used subsequently under more difficult vehicle driving conditions; (b) engine power control by cutting out cylinders or one complete engine—if two engines are provided; (c) use of a blower.

DIESEL ENGINES.

The wider utilisation of Diesel engines for vehicle propulsion requires considerable research work and design development. Very important problems are attached to engine reliability and ease of starting.

FUELS FOR MOTOR VEHICLE ENGINES.

In view of the great increase to be expected in the number of cars, tractors, and aircraft, it is feared that there will not be enough petroleum fuel. Besides, there are other reasons for not relying entirely on fuels made from petroleum. Considerable attention should be given to gas producer vehicles. So far wood blocks have been used exclusively. In future wide use should also be made of charcoal, coal, peat, semicoke, briquets made of agricultural waste, etc.

In this connection the following research problems require to be solved: (1) increasing the output of the producer gas engine to the level of the petrol engine; (2) improving gas producer design, including the gas purifier; (3) improving the methods of fuel preparation.

Other substitute fuels to be considered are natural gas, and gas made from coal and other substances.

In connection with the use of liquid substitute fuels it will be necessary to carry out research work to establish the requirements to be met by such fuels, and minor engine modifications to facilitate their economical use.

Further possibilities to be explored are battery driven vehicles, direct conversion of chemical energy into electrical, and atomic energy.

Engine Reliability.

Reliability is of great importance under the conditions of utilizations met with in the U.S.S.R. In opposition to foreign practice the introduction of standard replacement parts is advocated. Extended road tests with wet cylinder liners are proposed. Interchangeable valve seats and bearing inserts should be adopted. Experimental work should soon be carried out with special coatings on frictional surfaces, and also with special surface heat treatments and surface finishes.

With respect to lubrication, experiments have shown that oil supply to the cylinder before starting reduces wear 6 to 7 fold. Wide research ought to be carried out on this subject, and on other aspects of lubrication.

Specific Engine Power.

Research work should be carried out to show how far the specific engine power can be raised without detriment to reliability and engine life.

Fuel and Oil Quality.

The main object of research will be to find characteristic properties for oil and fuel which will give the best indication of actual performance in car engines on the road.

II. TRANSMISSION AND CHASSIS.

Clutch.

To reduce lining wear, a design should be developed in which masses would be so distributed as to accumulate heat and thus reduce heating of the lining to a minimum. Semi-centrifugal and automatic centrifugal clutches ought to be examined as these types are likely to ease de-clutching. In view of the great difficulty of mathematically examining the action of vibration damping springs, a special experimental set-up should be made to enable the action of such springs to be studied.

Change Speed Gear.

The chief object of research work should be increasing the ease of gear change, and automatic gearboxes should be included. It is considered that the automatic gearbox may greatly increase engine economy. The engine can operate at its most economical speed range and theoretical examination has shown that fuel economy may reach 25-30 per cent. Research work on Gear Teeth has not yet given a satisfactory answer and considerable further research is required.

Final Drive.

Tests carried out by the author have shown that cam type differential gears improve the performance of the automobile, especially on slippery road surfaces. Other experiments indicate that between the driving axles a differential gear is absolutely essential, or it must at least be possible to disconnect one of the axles, in order to eliminate "closed cycle" action. With respect to design, experiments should be continued on the effect of housing and bearing rigidity on axle drive wear.

Steering, Suspension and Frame.

There is wide scope for the introduction of hydraulic and pneumatic brakes. Tests should be carried out to find the optimum angle subtended by the brake lining, for theoretical considerations have shown that the angle used at present is not satisfactory as far as wear and ease of braking are concerned.

Methods should be developed which will enable direct determination of stresses in parts of the frame, the axles and the springs. Theoretical and practical investigations should be carried out to establish the effect of the angle of inclination of the steering knuckle on road performance of a car and on the ease of steering.

The trend towards softer springing makes necessary tests on lateral stability, including the application of the gyroscope principle.

Design, material and method of production have a great effect on tyre performance It has been shown that, under equal conditions, the coefficient of rolling resistance of a wheel varied between 1/3 and 1/3·5, the coefficient of friction between 1/2 and 1/2·5, etc. Research should be carried out by the motor car and tyre industry.

Improved Reliability with Reduced Dimensions and Weight.

Extensive research should be made towards improving the durability and wear resistance of all motor vehicle components and towards reducing their weight and dimensions. As a means towards achieving these objectives, new materials, new technological processes, etc., will have to be investigated.

Replaceable bushes and other parts should be widely applied in the design of Russian motor cars, so as to save wearing out expensive components.

Considerably more research work is required in order to enable the most suitable bearings to be chosen for the different parts of the vehicle, and to increase bearing life.

Lubrication.

The lubrication of chassis components is usually primitive and quite unsatisfactory. Even gearbox and differential are usually only splash lubricated, a method which fails with low quality oils and in winter. The position is worse still with respect to other chassis components.

A great deal of experimental work is necessary concerning: protection from dust and dirt; application of porous or graphited bushes that need no oil supply; introduction of improved lubricating appliances, etc.

Furthermore, the quality of the lubricating oils available for motor vehicles in U.S.S.R. is far from satisfactory and requires improvement.

Theory of the Automobile.

One of the scientific research subjects of foremost importance is the development of the theory of the automobile, including establishing a sound relation between design and performance.

The study of the following problems is of imminent importance: (1) fuel economy; (2) dynamics—ability to maintain a maximum mean speed under given conditions; (3) stability—ability to travel at the highest possible speed without skidding or overturning; (4) road performance—ability to travel on bad roads and cross-country; (5) ease of steering—with a minimum of effort and skill; (6) comfort—soft suspension, absence of noise and vibration.

While studying the performance of every single mechanism and determining more accurately the stresses on the components of these mechanisms, research work should be directed along the following lines: (1) examination of the functions of each mechanism and establishing criteria for judging the fulfillment of this function; (2) studying the kinematics and dynamics of the mechanism; (3) development of a method of determining, by experiment, the loads acting upon the parts of the mechanism; (4) development of a method of improved accuracy for designing individual components for rigidity and durability; (5) proper choice of metals for individual parts.

VIBRATIONS OF MACHINE FOUNDATIONS

By A. TOMASEK. (From *Strojnicky Obzor*, Vol. 26, Nos. 20 and 21, 20th October and 5th November, 1946, pp. 215-218 and 231-233, 10 illustrations.)

THE present reconstruction and reconversion drive throughout the industry gives increased importance to the problems of the most economical and suitable foundation of machines. The paper is devoted to pointing out the practical possibilities and the principles underlying the computation of " perfect " foundations.

FREQUENCY OF THE FREE OSCILLATION OF THE FOUNDATION.

The foundation with the machine resting on the floor or any other elastic material represents a mass separated from its surroundings by springs (see Fig. 1). The spring here is the equivalent of the elasticity of the bedding, or seating, whose mass can be neglected. The condition that has to be satisfied for the foundation to settle down uniformly under its own weight is that its mass centre and that of the elastic element (known as the centre of elasticity), O, in which the vertical resultant of the forces produced by the compression of the foundation acts, lie on a common vertical axis Z.

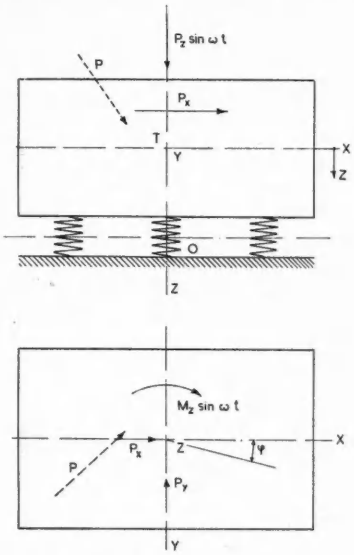

Fig. 1. **Principle of a vibrating foundation.**

The foundation with the machine be erected symmetrically to a vertical plane (e.g. XZ) which is also the plane of symmetry of the seating. The resultant of forces, whatever its magnitude and direction, is suitably resolved into a component P_z, a moment M_z, and components P_x and P_y in the XZ— and YZ— planes, respectively. Each of these produces only one definite kind of oscillation, so that the frequency of the free oscillation and the amplitudes of the vibration of the fundament can be computed separately.

VERTICAL OSCILLATIONS.

From the condition of the dynamic equilibrium of the foundation, acted upon by a periodic vertical force $P_z \sin \omega t$, we get the differential equation of motion in the Z-direction, viz :

$$m\frac{d^2z}{dt^2} + k_z\frac{dz}{dt} + c_z z = P_z \sin \omega t \quad .. \quad (1)$$

where k_z is the damping force of the seating, produced by unit velocity of the foundation, c_z the force to produce unit linear compression, and m the mass of the foundation. As the damping is mostly negligible, we can obtain the displacement of the foundation from equation (1)

$$z = z_o \sin \omega t \quad .. \quad .. \quad (2)$$

By substituting we get the amplitude of the oscillations

$$z_o = \frac{P_z}{c_z - m\omega^2} \quad .. \quad .. \quad (3)$$

From this we conclude that resonance of the exciting frequency with the frequency of the free oscillations of the foundation occurs for

$$c_z = m\omega^2 \quad .. \quad .. \quad .. \quad (4)$$

the frequency of the free oscillation (natural frequency) being,

$$\omega_v = \sqrt{\frac{c_z}{m}} \quad .. \quad .. \quad .. \quad (5)$$

which is seen to depend on the mass of the foundation and elasticity of the seating. Further substitutions yield an expression for the amplitude in terms of exciting force and frequencies,

$$z_o = \frac{P_z}{m(\omega_v - \omega^2)} = \frac{P_z}{m\omega_v^2} \cdot \frac{1}{\left(1 - \frac{\omega}{\omega_v}\right)^2} = \frac{P_z}{m\omega_v^2}\,\nu \quad (6)$$

The amplitude of the force transmitted to the surroundings is then expressed by the relation

$$P_{zo} = c_z z_o = m\omega_v^2 z_o = P_z \nu \quad .. \quad (7)$$

where the constant ν is a known dynamic coefficient (transmittivity). The plot of this coefficient against the ratio ω/ω_v is seen in Fig. 2. It represents the fraction of the exciting force transmitted to the surroundings, and is an important criterion for the suitability of the seating. The smaller it is, the more appropriate is the seating. Fig. 2 also gives a curve showing the effect of damping. Comparison of these two curves shows that outside the zone of resonance, damping can always be neglected in practical calculations. The dotted curve in Fig. 2 gives the ratio of z_o, the actual displacement of the foundation, to $z_o' = P_z/m\omega^2$, or the value

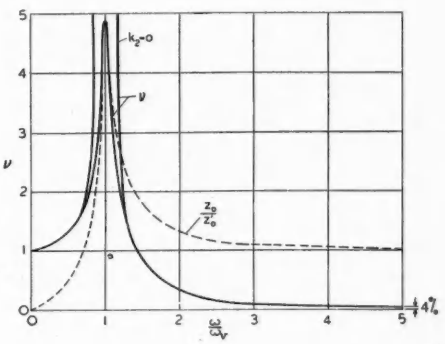

Fig. 2. **Dynamic coefficient (transmittivity) and ratio of amplitudes.**

$$\frac{z_o}{z_o'} = \left(\frac{\omega}{\omega_e}\right)^2 \nu,$$

where z_o' is the displacement of the same foundation assumed to be free in space, equivalent to the case of a foundation mounted on springs of an infinitely small elastic constant.

It should be noted that Fig. 2 represents the case of constant amplitude of the exciting force, i.e. of an engine running at constant speed. By varying the natural frequency ω_e of the foundation we tried to find the most suitable mounting. But, if we assume a foundation mounted on springs with a given elastic constant, our object being to find the displacement of the foundation at various engine speeds, we must bear in mind that the exciting force varies with the square of the speed, or frequency of the engine.

OSCILLATIONS ABOUT A VERTICAL AXIS.

The moment $M_z \sin\omega t$ produces only a rotation about the Z-axis. The analysis of this case will proceed along lines analogous to those under the previous heading, provided that the mass m be replaced by the moment of inertia of the foundation with respect to the Z-axis, $\mathcal{J}_z$, and the elastic constant, c_z, by the elastic moment of the mounting with respect to rotation about the Z-axis, d_z. This is the moment produced by rotating the foundation about Z through 1 radian.

The equation of motion then takes the form

$$\mathcal{J}_z \frac{d^2\varphi}{dt^2} + d_z \frac{d\varphi}{dt} = M_z \sin \omega t \quad .. \quad (8)$$

and yields a natural frequency

$$\omega_t^2 = \frac{d_z}{\mathcal{J}_z} \quad .. \quad .. \quad .. \quad (9)$$

The angular displacement ratio φ_o/φ_o' depends on the ratio ω/ω_t.

OSCILLATIONS IN THE XZ AND YZ PLANES.

Assuming symmetry of the foundation about the XZ or YZ plane, the oscillations in these two planes are independent of each other, and can be produced only by forces and moments acting in the respective plane. The vertical component has already been dealt with; only the horizontal components are left to be considered.

Referring to Fig. 3, let the position of a foundation, displaced by a moment $M_y \sin \omega t$ and a force $P_x \sin \omega t$, be determined by a displacement of the mass centre, x, and an angular displacement about this centre, φ_y. Further, let the resistance of a single spring against displacement along the X-axis be c_{xi}, and in the vertical

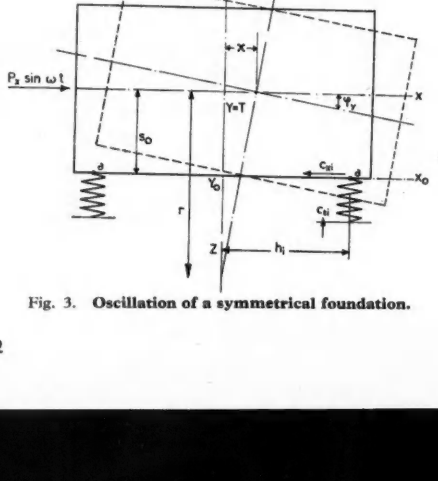

Fig. 3. **Oscillation of a symmetrical foundation.**

direction, c_{zi}. Supposing the springs to be supported in points a, all these reactions will pass through the said points. Assuming all springs to be similar and arranged symmetrically about Z, then mass centre and centre of elasticity will lie on the common vertical axis Z.

The resistance, or reaction, of the whole mounting (consisting of n springs) is produced by the displacement of the foundation along X of a magnitude x,

$$-x \,.\, nc_{xi} = -c_x \,.\, x$$

and the moment of all springs about the mass centre, produced by a deflection of the foundation about the mass centre through an angle φ_y

$$-\varphi_y \sum_1^n h_i^2 \, c_{zi} - \varphi_y \, s^2 \, c_x = -c_y \, \varphi_y \; .. \; (10)$$

Apart from this, the displacement results in a moment about the mass centre of the foundation, $+xc_x s$, and again, as a consequence of the rotation of the foundation through an angle φ_y, in a force along the axis, $+\varphi_y c_x s$. The equations of motion of the foundation then become

$$\left.\begin{aligned} m\frac{d^2x}{dt^2} + c_x \,.\, x - c_x \, s \, \varphi_y &= P_x \sin \omega t \\ \mathcal{J}_y \frac{d^2\varphi}{dt^2} + d_y \varphi_y - c_x \, sx &= M_y \sin \omega t \end{aligned}\right\} \quad (11)$$

The particular integrals of these equations representing the displacements will be

$$\left.\begin{aligned} x &= x_o \sin \omega t \\ \varphi_y &= \varphi_{yo} \sin \omega t \end{aligned}\right\} \quad .. \quad (12)$$

By substituting into equations (11) it follows

$$\left.\begin{aligned} x_o &= \frac{(d_y - \mathcal{J}_y \omega^2) \, P_x + c_x s \, M_y}{(c_x - m\omega^2)(d_y - \mathcal{J}_y \omega^2) - c_x^2 s^2} \\ \varphi_{yo} &= \frac{c_x s \, P_x + (c_x - m\omega^2) \, M_y}{(c_x - m\omega^2)(d_y - \mathcal{J}_y \omega^2) - c_x^2 s^2} \end{aligned}\right\} \quad (13)$$

The force in the mass centre, P_x', and the moment M_y' transmitted to the surroundings are then

$$\left.\begin{aligned} P_x' &= c_x x_o - c_x s \, \varphi_{yo} \\ M_y' &= d_y \, \varphi_{yo} - c_x s \, x_o \end{aligned}\right\} \quad .. \quad (14)$$

The natural frequency can be determined, as before, from the condition that the amplitudes x_o and φ_{yo} are infinite. By equating to zero the common denominator of the fractions in the equations (13) we get an equation of the 4th order from which results

$$\omega_{12}^2 = \frac{1}{2m\mathcal{J}_y}[c_x \mathcal{J}_y + d_y m \pm \pm \sqrt{(c_x \mathcal{J}_y + d_y m)^2 - 4(d_y c_x - c_x^2 s^2)}] \; .. \; (15)$$

It is seen that two frequencies are obtained corresponding to rotations of the foundation about two points on the Z-axis whose distance can be calculated from the equations (13), viz.

$$r = \frac{x_o}{\varphi_{yo}}$$

Similar expressions hold for the YZ-plane.

These theoretical considerations show the principles to be followed in erecting foundations, namely :

(1). Resonance, i.e. coincidence of the exciting frequencies with the natural frequencies of the foundation must be avoided.

(2). The softest mounting should be chosen in order to keep all frequencies of the oscillations as far below the exciting frequency as possible ; for it follows from Fig. 2 that the greater this difference (e.g. $\omega_v < \omega/3$) the smaller the force transmitted to the surroundings. We see, however, that in this case the displacements will no longer be insignificant. Undue increases of displacements should be prevented by increasing the mass and moment of inertia of the foundation. This condition of operation is known as the super-critical operation; in running-up the machine (or engine) the resonance zone has to be traversed very rapidly.

(3). Where, for any reason, super-critical operation is impossible, a very hard, or rigid, mounting must be chosen, and the foundation should be made as light as possible. This has the effect of raising the natural frequency high above the exciting frequency, $\omega/\omega_v \leqslant 1$. As we cannot prevent the whole of the exciting force or moment from being transmitted, at least we can, by these means, reduced the amplitude of the displacement of the foundation.

TYPES OF, AND MATERIALS FOR MOUNTINGS.

(1). The simplest and cheapest method of bedding is to set the foundation on natural or artificially densified ground. The resilience of the ground, usually characterised by its elasticity constant in a vertical direction, c, and in the horizontal direction, S, and expressed in kg./cm.^{3}, is the stress in kg./cm.^{2}, set up by unit extension or compression of the foundation. These constants depend on the nature and specific pressure of the ground, and often on the exciting frequency and the amplitude of the displacement of the foundation. They are determined by means of vibrators.

At current values of the specific pressure of 0·5 to 0·8 kg./cm.^{2}, these values are roughly as follows,

Fine Sand	..	$C = 5 - 10$, $S = 4 - 5$ kg./cm.^{2}
Coarse Sand	..	$C = 10 - 20$, $S = 5 - 6$ „
Dry Loam	..	$C = 13 - 20$, $S = 4 - 5$ „

Natural frequencies of foundations seated on the ground usually lie between 800 and 3,000 cycles per minute.

Where the ground consists of incoherent layers it must be strengthened by driving down pilots, a method also used where the ground is too resilient.

(2). The cheapest and least satisfactory material for artificial seatings is cork, made of cork waste bounded by iron hoops (armed cork). Apart from this, soft cork is also used, this mainly as lateral covering and sound insulation. Where cork seatings are used, the whole foundation area has to be laid under with cork because of its low compression strength. Felt is of high, but quickly disappearing elasticity. Artificial products of animal hair ("mascolite") is used in the U.S.A.

(3). Rubber seatings were preferred until quite recently, mainly for being softer and much stronger than cork, and on account of their considerable damping capacity. Massive or hollow rubber bungs are used singly and in pairs or sets, between cast iron or steel plates. Their modulus of elasticity, depending on the hardness of the rubber, varies between 9 and 50 kg./cm.^{2}, on impact loading, but has only about one third of these values for static or alternating loading. As rubber may easily be vulcanised, or bonded to metals, it lends itself to being built into insulators of the type shown in Fig. 4.

(4). At present helical springs are used almost exclusively. An example is given in Fig. 5. By tightening or loosening the adjustable screw of individual insulators not only may the foundation be correctly positioned, but

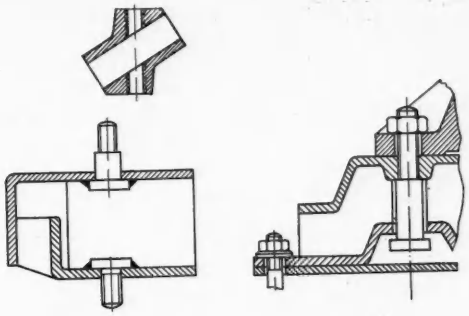

Fig. 4. **Special insulator of the vulcanized-on (bonded) type.**

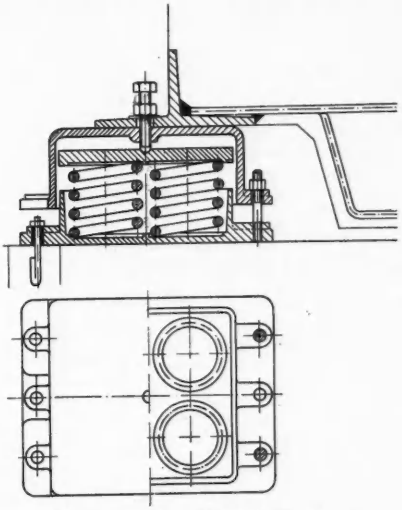

Fig. 5. **Insulator with helical springs.**

inaccuracies in the execution of the foundations may be compensated for, and uniform loading of all the insulators obtained.

To give free access to the adjusting screws, the usual arrangement of the insulators is alongside the foundation by means of T or U-shaped irons.

A drawback of this arrangement generally is its low stability, especially in the case of drop-hammers and similar high-built machines. Therefore, for such machinery, the suspension system is preferred (see Fig. 6). This system is also used with advantage for horizontal low-speed combustion engines, where the exciting forces act mainly in a horizontal plane. As the natural frequency is approximately $\omega_v^2 = \frac{g}{l}$ (pendulum of length l), it is obvious that the frequency becomes

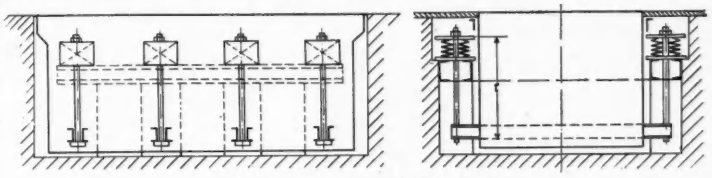

Fig. 6. **Spring-suspended foundation.**

very low when l is considerable. At an engine speed of, say 120 r.p.m., this assures very favourable operation (super-critical) of the engine in the horizontal plane. From Fig. 2 then follows that only about 4 per cent of the exciting forces are transmitted to the surroundings. The horizontal considerable displacement has to be reduced by a heavy foundation. In the vertical direction, sub-critical operation will be permissible, as the forces are negligible, although they are fully transmitted to the surroundings. The spring rate in a vertical direction will be such as to keep the natural frequency between limits corresponding to 300 to 400 r.p.m.

It should be noted that a suitable design of the foundation and mounting also reduces the friction losses due to engine vibrations, and thus recovers between 5 and 10 per cent of the engine output.

FRANCE

THE INFINITELY VARIABLE SPEED GEAR "SADIVAR"

(From *Le Génie Civil*, Vol. 123, No. 22, November 15, 1946, pp. 315-316, 2 illustrations.)

THE "Sadivar" gear, a sectional arrangement of which is shown in Fig. 1, transforms a constant motor speed into a driven shaft speed which is variable at will in infinitely small steps and may be higher or lower than the motor speed. It consists in a form of thrust ball bearing where the balls have been replaced by three friction wheels a (two only are shown) whose axes b converge in a common articulation point c which can move freely. On the outside, the three axes slide in slots d of the housing so that their relative angles of 120 deg. are always correctly maintained. The motor is coupled to shaft e on to which is keyed the disk M with its hollow-ground, torus-shaped groove to accommodate the friction wheels a. Another disk R with an identical torus-shaped groove is driven by the wheels. If the wheels roll on radius r_1 of disk M and drive disk R on radius r_2, then the speed of output shaft s is given by $v_2 = v_1 \cdot r_1/r_2$, where v_1 is the constant speed of motor shaft e. A change of inclination of the wheel axes causes a variation of diameters r_1 and r_2 and therefore of output speed. For r_1 equal r_2, motor and output speed are identical. If in one end position of the axes, the driving radius r_1 is one third of the driven radius r_2, then the output speed will be one third of the motor speed. In the other end position, the output speed will be three times the motor speed. Thus, the total theoretical output speed ratio will be 1 in 9; in practice, however, the maximum ratio will be 1 in 7.

The arrangement of three friction wheels combines perfect static equilibrium of the gear with a high transmissible torque. The disks M and R and the wheels are made of fully hardened steel and are ground to true shape. The mechanism runs in oil and lubrication is assured by provision of an oil pump. Correct rolling action without sliding is only possible if the rolling radii r_1 and r_2 are exactly the same for all three

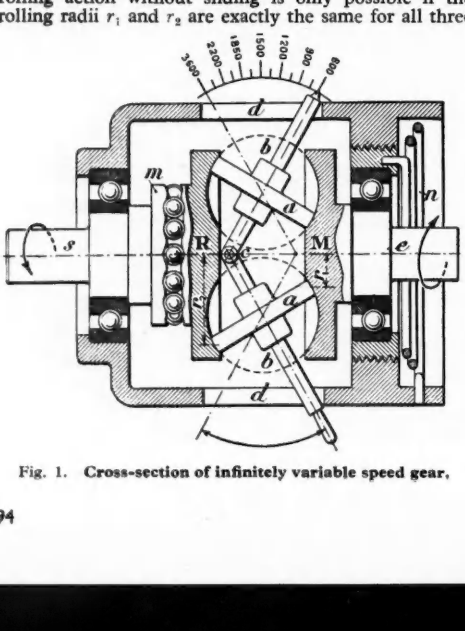

Fig. 1. **Cross-section of infinitely variable speed gear.**

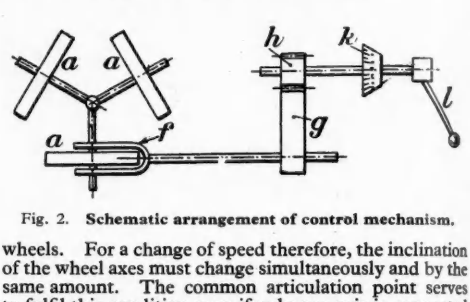

Fig. 2. **Schematic arrangement of control mechanism.**

wheels. For a change of speed therefore, the inclination of the wheel axes must change simultaneously and by the same amount. The common articulation point serves to fulfil this condition even if only one axis is connected to the control mechanism. A gyro effect tends to stabilize the position of the axes. The speed control is shown schematically in Fig. 2. A fork f operated by gear segment g turns one spindle a and thus changes the inclination of the three connected spindles. Segment g is driven by pinion h which is operated by hand lever l. A graduated cone k on shaft h-l indicates the amount of speed increase or reduction. An automatic blocking device retains the control in any desired position.

AUTOMATIC CONTROL OF SURFACE PRESSURE.

The surface pressure should theoretically be proportional to the torque which is transmitted by friction. This is obtained by the arrangement of a "thrust bearing with sloping races" (m in Fig. 1) which connects disk R to output shaft s. The races of this bearing are sloping so that each ball is retained in a groove. When, due to rotation of R, the balls begin to roll, they tend to climb up the slopes thus exerting a thrust force on the friction wheels. The surface pressure is therefore variable and the losses through slipping of the wheels are kept small. Mechanical efficiency is maintained at a high level for the whole torque range. Surface wear is insignificant as the rubbing surfaces are made of hardened steel. After a long running time, however, the slopes of the thrust races m will wear down and reduce the effect of pressure adjustment. As a remedy, the ball bearing supporting driving disk M is fitted into a screwed cover. Whenever the machanism is stopped, the torsion spring n which is arranged to act on the screwed cover, tends to re-establish surface contact between disks and wheels and to exert a small surface pressure.

The Sadivar mechanism is balanced statically by the use of three wheels, dynamically by the common articulation of the three axes. This automatically ensures a symmetrical position of the wheels and correct rolling action. The mechanism can be built as a simple variable gear with co-axial driving and driven shafts, as a variable motor gear with flange-fitted motor, as a variable reduction gear, or finally as a variable motor reduction gear with the mechanism fitted on flanges between motor and reduction gear.

SNOW FLAKES AND HAIRLINE CRACKS IN WELD METAL

By M. LEFÈVRE. (From *Revue de la Soudure*, Vol. 1, No. 2, 1945, pp. 39-49, and *Arcos*, Vol. 23, No. 102, July, 1946, pp. 2411-2429, 14 illustrations.)

DESCRIPTION OF DEFECT.

SNOW flakes and hairline cracks appear in steels and weld metal but their effects are more obvious in steels than in weld deposits. The defects do not occur in steels of normal structural quality and their investigation started only fairly recently, with the development of low-alloy steels during the 1914/18 war and with the increasing use of high-quality welding electrodes since 1938/39. Polished and etched cross-sections of certain steels show hairline cracks as a system of interconnected, or isolated, fine fissures (Fig. 1a). Fractured tensile test pieces of these metals contain silvery, round or oblong, patches of a grained texture ("snow flakes," Fig. 1b) against the darker streaked background of the fracture. Snow flakes do not necessarily develop from hairline cracks but may form during the tension test in spots which have shown no cracks before test or may even appear on test pieces showing no visible hairline cracks at all. The cracks may be very fine and widely distributed, or coarse and isolated, and, in this case, they start out from different defects like cavities and inclusions. The steel shown in Fig. 1 is obviously weakened by extensive defects but normally the appearance of snow flakes coincides with only an insignificant drop in ultimate tensile strength whilst the deformability, i.e. elongation and area reduction, decreases by 50 per cent or more. Normally, weld deposits show the defect in form of snow flakes only.

CONDITIONS CONDUCTIVE TO APPEARANCE OF DEFECT.

Low-alloy steels are susceptible to the formation of snow flakes, steel analysis however, is no safe guide as to the probability of their appearance. Crucible steels do not show the defect but susceptibility increases in the following order: acid open-hearth steel, acid electric steel, basic open-hearth steel, basic electric steel.[1] Rolled and especially forged pieces are the more susceptible the quicker they are cooled and the larger their mass.

Snow flakes also appear in weld deposits from heavily coated electrodes especially if the coating contains basic or organic constituents. The defect occurs more frequently in high-resistance welding and is more wide-spread in welds deposited at a relatively low temperature. It is never found in deposits rich in oxygen (bare electrodes or electrodes with coatings rich in iron oxide). The defect is therefore peculiar to high-quality, well deoxidized welds. Even if the tensile fracture of a weld metal shows many snow flakes, no previous hairline cracks are discernible by radiographic, magnetoscopic, or metallographic and microscopic examination, nor can physical or structural abnormalities be detected before test. Snow flakes only appear in fractures of certain types of tests (mainly tensile), are rare in others (e.g. bending) and never found after impact tests. They develop in a direction perpendicular to the tension load whatever their direction in relation to the welding bead, and they can be made to disappear by thermal treatment even at low temperatures.

The formation of snow flakes has been studied on polished test cross-sections under the microscope.[2, 3] Before test, only the normal inclusions and impurities can be seen. Under increasing tension, the metal is first deformed round the inclusions, then a crack appears in a direction transverse to the tensile load, finally deformation increases around the crack until rupture occurs. The fracture shows the known snow flake aspect at the exact spot where the crack had first been visible (see Fig. 2).

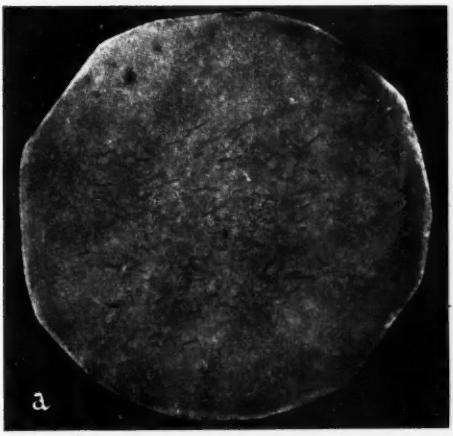

Fig. 1. **Snow flakes and hairline cracks in steels.**
(a) In a cross-section, (b) In the fracture.

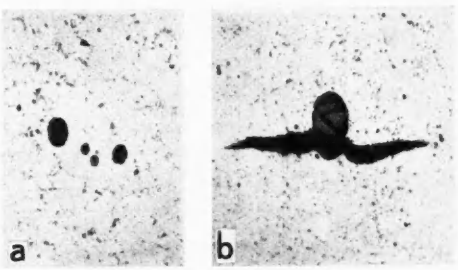

Fig. 2. **Snow flake formation in weld metal.**
(a) Metal deformation round an inclusion.
(b) Appearance of crack in transverse direction to tension stress.

ORIGIN OF DEFECT.

(a) *Hydrogen content.*

Causes for the formation of hairline cracks and snow flakes were first thought to be internal stresses (unequal contraction, hot working, allotropic changes), metallurgical defects (inclusions, segregations, gas inclusions), and, in welds, the differential ductility of annealed underlying layers and rapidly air-cooled top runs.[4] The main cause, however, has been found to be the hydrogen content of the steel,[5] other circumstances being contributory if hydrogen is present in sufficient quantity. The hydrogen content of steel is difficult to determine because atomic hydrogen rapidly diffuses out of the steel.[7] Also, it is not easy to decide in what form hydrogen is present in the steel and whether all of it

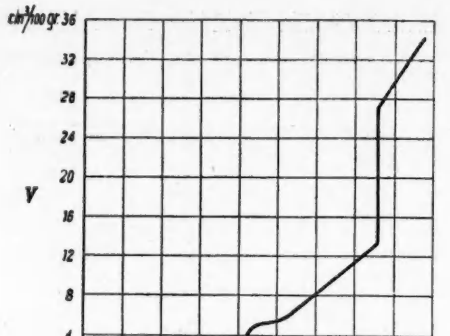

Fig. 3. **Hydrogen solubility in iron (Sieverts).**
Abscissa : Temperature T in deg. C.
Ordinate : Volume V of hydrogen dissolved in iron.
(1 cu. cm./100gr. = 0·28 cu. in./lb.).

has been extracted during analysis. Its effect on steel has therefore usually been investigated on test pieces progressively charged with increasing amounts of hydrogen.[6]

Fig. 3 gives the solubility of atomic hydrogen in iron and Fig. 4 the permeability for atomic hydrogen of a Cr-Ni-W steel. Molecular hydrogen is not soluble in iron and cannot diffuse into, or out of, the steel. According to Fig. 3, considerable quantities of atomic hydrogen are liberated during solidification and cooling, and they diffuse either out of the steel or into internal cavities, segregations, or cracks. There they recombine to molecular hydrogen whose gas pressure rises until it balances the atomic hydrogen pressure in the steel. On slow cooling, most of the hydrogen has time to diffuse out of the steel through the steel surface but on quenching, steel permeability decreases so rapidly that nearly all the hydrogen remains in the steel and the molecular hydrogen pressure in internal cavities may reach high values. This pressure has been determined theoretically and may, even for normal hydrogen percentages, reach and surpass the tensile strength of the steel or weld, provided no hydrogen diffuses out of the steel. Thus, a rapidly quenched hydrogen-rich steel may in itself show hairline cracks but normally these defects appear only if other factors are present (internal or external tension). Under suitable conditions, the internal hydrogen pressure may reach a value only slightly below the fracturing limit of the steel and a small external tension, or a sudden temperature change, may then suffice to produce cracks as in Fig. 1a, and in the fracture, snow flakes as shown in Fig. 1b. If internal pressure remains low, then external tension must raise the stress beyond the elastic limit before cracks can appear (Fig. 2). In this case, the defect is only visible after test, in form of snow flakes in the fracture (where localised stresses were highest), or in form of hairline cracks in the neighbourhood of the fracture. Snow flakes, therefore, appear when the hydrogen content is relatively low, and other factors largely contribute to their formation. Hairline cracks, however, may appear before test or under small external influences if the hydrogen content is high enough, i.e. considerably higher than for snow flake formation (see Fig. 5). In weld metal, the defects are usually less serious than in steel.

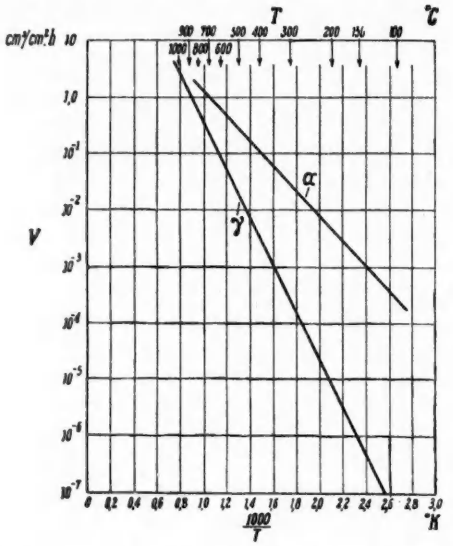

Fig. 4. **Permeability of Cr-Ni-W steel in α and γ states (Bennek).**
Abscissa : Temperature T in deg. C. and inverse of temperature, 1000/T in deg. abs.
Ordinate : Volume of H_2 which passes through 1 cm² of 1 mm. thick steel in 1 hour.

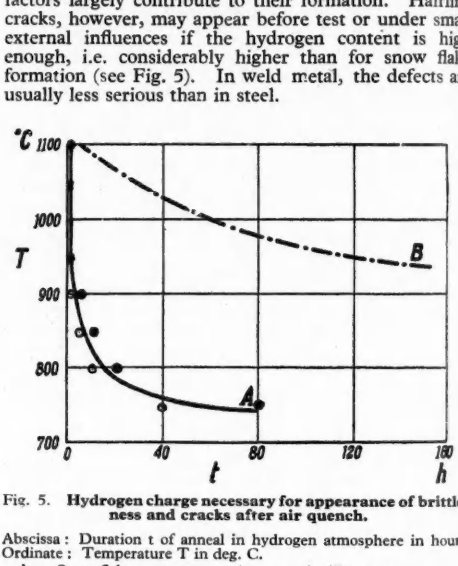

Fig. 5. **Hydrogen charge necessary for appearance of brittleness and cracks after air quench.**

Abscissa : Duration t of anneal in hydrogen atmosphere in hours.
Ordinate : Temperature T in deg. C.
A = Snow flake appearance. O = metal still sound.
● = Reduction of elongation and appearance of snow flakes.
B = Appearance of hairline cracks.

During the tensile test, cracks begin to form in zones of high local stress concentration (around impurities) but as soon as a cavity has formed, the hydrogen pressure falls, the cavity is relieved of stress and the adjacent sound zones are deformed and finally fractured during the normal progress of the test. Hence the typical snow flake aspect of the fracture shown in Fig. 1b. No snow flakes appear in impact tests as impact cracks start at the root of the notch and not in local hydrogen-loaded weak spots. The minimum hydrogen content liable to produce snow flakes varies widely and also depends on external factors like quenching temperature, cooling rate, mechanical stresses, and chemical composition. A few test results may give useful indications :

Ordinary mild steel, no snow flakes, ..	½ to 1 cu. in. H_2/lb.
C-steel (hydrogen charged) no snow flakes,	2·5 to 3·5 ,,
Ni-Cr Steel, showing snow flakes	2 to 3·5 ,,
Weld deposit, no snow flakes	½ to 1 ,,
Weld, 28·5 tons p. sq. in., hardly any snow flakes	6 ,,
Weld, 32 tons p. sq. in., a few snow flakes ..	1·6 to 2·8 ,,
Weld, 35/38 tons p. sq. in., many snow flakes	5 to 8·5 ,,

(b) *Influence of cooling rate.*

The more slowly a steel is cooled the fewer snow flakes appear in the fracture of tensile test pieces. Below a certain critical cooling rate (depending upon the chemical composition of steel and upon other factors) snow flakes disappear completely.[6] Tests on weld deposits from various electrode types (tensile test pieces 20 × 15 × 200 mm. ($\frac{13}{16}$ × $\frac{8}{8}$ × $7\frac{7}{8}$ in.), cut from weld metal susceptible to snow flake formation) gave additional information: Taking the total snow flake area within the fracture as an empirical value of the amount of snow flake incidence, the author could show that oil, hot water, or air-stream quenched test pieces are even more susceptible to snow flake formation than cold-water quenched pieces.

Cold-water quench: snow flake incidence	1·7	average
Oil quench: snow flake incidence	4·5	,,
Hot-water quench: snow flake incidence	5·4	,,
Air-stream quench: snow flake incidence	5·2	,,
Still-air quench: snow flake incidence	2·3	,,
Kieselgur or furnace cooling: incidence	0·0	,,

The result of these tests has been confirmed[5] and it can be assumed that with extremely high cooling rates, the hydrogen has no time to segregate, remains in solution and cannot cause that internal pressure which leads to crack formation. In tests on steel[6] which did not show this reduction of snow flake formation in cold-water quenched pieces, the greater mass of the test pieces chosen may have reduced the attainable cooling rate to that of an oil-quench. In subsequent tests on welds, air-stream cooling was adopted wherever a great amount of snow flake incidence was desired.

(c) *Influence of quenching temperature.*

Snow flake formation in steels diminishes with falling quenching temperature. Certain steels no longer show snow flakes when the quenching temperature drops below 600 deg. C., others contain snow flakes when quenched from 200 or even 100 deg. C. In these, the flakes must form in a temperature range between 200 and 20 deg. C. In weld deposits the incidence of snow flakes decreases rapidly if the quenching temperature drops from 900 to 600 deg. C. A rapid quench from 500 deg. C. does not produce any snow flakes (see Fig. 6). This proves that snow flakes can be found if a weld, annealed above 600 deg. C., is not cooled slowly enough.

(d) *Influence of tensile stresses.*

Snow flakes only appear when to the internal hydrogen pressure stresses are added thermal or mechanical stresses due to unequal contraction, allotropic volume changes (the more efficacious the lower their temperature), hot working, or external tension. As these stresses cannot be measured, no quantitative relationship between stresses, hydrogen content and snow flake incidence can be established.

(e) *Influence of chemical composition.*

Certain steels never show snow flakes, others are always susceptible to their formation. Steels in order of increasing susceptibility are[6]: *C*-steels, 1 to 1·5 per cent; *Mn*-steels, *Mn-Si* steels, 1 to 2 per cent; *Cr*-steels with traces of *Mo* or *Va*, 1·5 per cent or higher; *Ni*-steels, *Cr-Ni* steels, *Cr-Ni-Mo* and *Cr-Ni-W* steels. On the other hand, ferritic *Cr*-steels, austenitic *Ni-Cr* or *Mn* steels, 4 per cent *Si*-steels and high-speed steels never show this defect. Among weld metals those with 1 to 1·5 per cent *Mn*, with *Si* or *Cr* (3 per cent) are highly susceptible. *Mo* in small amounts has no influence.

Some additions favour hydrogen absorption (like *Si*), others influence the widely varying steel permeability, others again depress the transformation point, thus reducing steel permeability when a great amount of hydrogen is segregated. The hydrogen cannot escape and the internal pressure rises.

TABLE I. HYDROGEN CONTENT OF CERTAIN STEELS.[3]

Metal	Hydrogen content (cu. in. per lb.)
Ordinary mild steel	0·25 to 0·32
Electrode wire	0·32 to 1·12
Weld deposit	1·7 to 8·4

TABLE II. INFLUENCE OF ELECTRODE COATING ON SNOW FLAKE FORMATION.

Coating	Hydrogen content in deposit. cu. in. per lb.	Appearance of snow flakes.
With 3·5 per cent organic matter	6·9	extensive
do. dried 2 hrs. at 400 deg. C.	3·7	restricted
Without organic matter	3·7	restricted

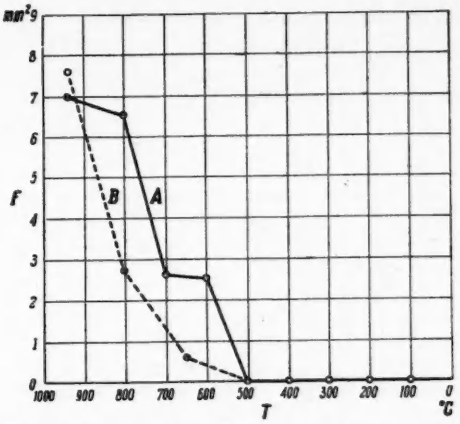

Fig. 6. **Influence of air quench temperature on snow flake formation.**

Abscissa: Temperature T of air quench, in deg. C.
Ordinate: Snow flake surface in fracture, in sq. mm.
A = Heating for ½ hour at 920 deg. C., slow cooling in furnace to the indicated temperature, then air quenching.
B = Heating for ½ hour at the indicated temperature, then air quenching.

COUNTER MEASURES.

(a) *Reduction of hydrogen absorption in the molten metal.*

Weld metal which is susceptible to snow flake formation contains 10 times more hydrogen than sound metal. Tables I and II show that this hydrogen does not come from the electrode wire (whose hydrogen content has been proved to be of no influence) but from organic constituents and the humidity of the coating, and that it can be reduced by elimination of organic matter or, to a small extent, by drying at 300 deg. C., and to a larger extent, by drying for 3 hours at 350 deg. C. This treatment is not sufficient for complete elimination of snow flakes, drying at higher temperatures however would spoil the coating. The humidity of the atmosphere is of little influence except insofar as it bears on the humidity of the coating itself.

The possibilities of preventing hydrogen absorption at the source are thus limited, and the hydrogen has to be extracted from the solidified metal. Extraction in vacuo has been tried, the technique, however, is still complicated and only applicable to small test pieces.

(b) *Thermal treatment.*

Heating of steel or weld and subsequent slow cooling reduce snow flake incidence for reasons explained before. Generally, in order to avoid safely any defects, it is recommended to cool the metal slowly from 900 deg. C., and, in particular, to cool the metal slowly through the critical range at 200 deg. C. Cooling should take 10 hours normally, and up to 100 hours

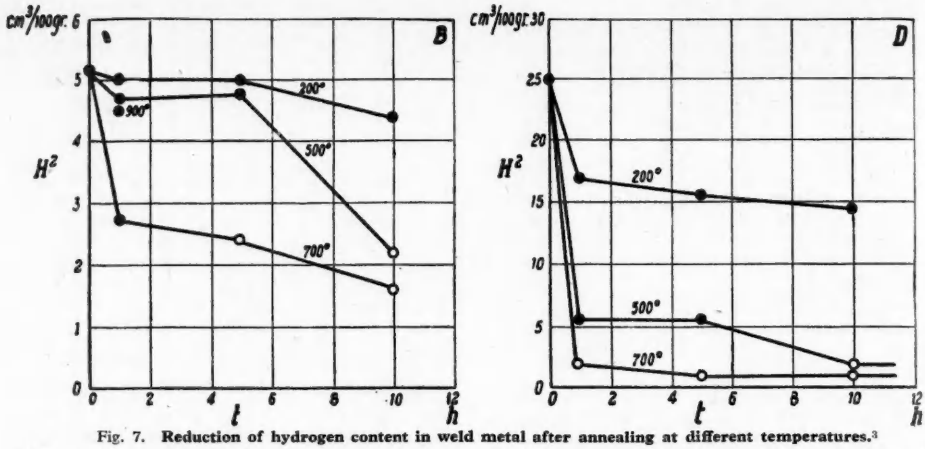

Fig. 7. **Reduction of hydrogen content in weld metal after annealing at different temperatures.**[3]
Abscissa: Annealing time t in hours. Ordinate: Hydrogen content in cu. cm. per 100 gr. (1 cu. cm./100 gr. = 0·28 cu. in./lb.).
B and D = Test electrodes.

in unfavourable cases. Various investigators found the following results on actual tests:

(a) Steel showing snow flakes in fractures and a noticeable reduction of elongation was restituted to its original healthy condition by heating at 100 deg. C. for 24 hours.[6 and 10]

(b) In welds, snow flakes and diminished ductility could be avoided by welding at temperatures above 150 deg. C. (pre-heating) and subsequent annealing at 600 deg. C.[4]

(c) Snow flakes in welds are avoided by heating to 230 deg. C. immediately after welding and maintaining this temperature for 8 hours.[11]

(d) Snow flakes in welds are avoided by heating at 600 or 500 deg. C. for 15 mins., at 400 deg. C. for 1 hour, at 300 deg. C. for 3 hours, at 200 deg. C. for 5 hours, or at 100 deg. C. for 50 hours, heating always to be followed by a cold water quench (see also Fig. 8). Even after leaving at ambient temperature for a very long (unspecified) time, an improvement could be noticed.[12]

(e) After short heating above 200 deg. C., or after several weeks at ambient temperature, steel bars showed a definite increase in ductility without change of elastic or ultimate limits.[13]

(f) By heating 1 hour at 700 deg. C. or 10 hours at 500 deg. C., the hydrogen content of weld metal could be so far reduced (approx. 0·5 cu. in. per lb.) that even a rapid quench did no longer lead to snow flake formation (see Fig. 7). Heating below 400 deg. C. had no pronounced effect as the hydrogen is then in its molecular state and cannot diffuse out of the weld metal. Heating above the transformation point of the steel had no effect because, in this case, the greater solubility of hydrogen in steel leads to retention of atomic hydrogen.[3]

Tests results are thus inconclusive and partly contradictory, and more systematic tests were undertaken by the author. Welds deposited from electrodes with coatings likely to lead to heavy snow flake formation in the deposit, were cut into tensile test pieces and quenched in an air stream in order to produce the maximum possible amount of snow flakes in the fracture. Some specimens were fractured in this condition, and the presence of snow flakes was ascertained. Other specimens, in batches of four, were then heated to a definite temperature, maintained at that temperature for a certain period and slowly cooled in the shut-down furnace. The test bars were then finish machined and fractured in tension. Table III gives the duration of heat treatment needed to obtain a fracture free of snow flakes, and graph Fig. 8 shows that this duration is an

TABLE III. DISAPPEARANCE OF SNOW FLAKES WITH TREATMENT AT DIFFERENT TEMPERATURES.

Temperature of treatment, deg. C.	Time needed for disappearance of snow flakes	Approx. average time in hours.
920	0	0
650	0	0
265	30 to 40 mins.	0·6
120	25 to 75 hours	50
room temperature	1 to 3 months	1000

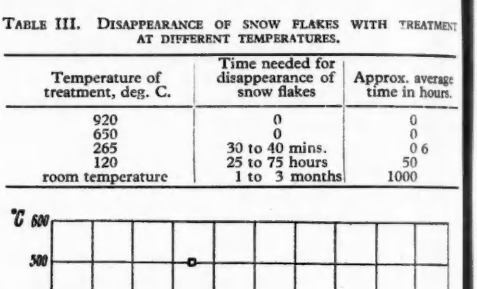

Fig. 8. **Snow flake suppression by low-temperature treatment, duration of treatment as a function of temperature.**
Abscissa: Duration t in hours needed for complete disappearance of snow flakes.
Ordinate: Annealing temperature T in deg. C.
○ = Author's tests (slow furnace cooling).
□ = Tests (d.)[12] (water quenching).
△ = Tests (c.)[11] (air quenching).

exponential function of the maximum temperature attained. Results also prove that snow flakes disappear without any thermal treatment, after a very long time. Tests (d) (indicated by small squares in Fig. 8) led to rather longer times of treatment but this can be explained by the rapid quench employed in those tests, as compared with slow furnace-cooling in the author's tests.

If test bars treated in this manner for elimination of snow flakes are again air-quenched, the snow flakes will reappear to the same extent as before the thermal treatment (see Table IV). This treatment, therefore, suppresses snow flake formation but leaves its causes in the metal. Hydrogen is not eliminated but rendered innocuous. On the other hand, test pieces annealed at 500 to 700 deg. C. for several hours (tests f) have had their hydrogen content so far reduced that even a

rigorous quench does not lead to further snow flake formation. One may conclude that the low temperature treatment which neutralises, but does not eliminate, hydrogen, is sufficient where the metal is not submitted to a later heat treatment including a rapid quench.

CONCLUSIONS.

(a) *Limitations of tests.*

The author's tests refer to weld metal and snow flake formation only. No deductions can be made for snow flake and hairline crack formation in steels where the effect of hydrogen is much more serious. As different electrodes give weld deposits of widely varying susceptibility for snow flakes the quantitative test results refer to test electrodes only. Qualitative deductions may however be drawn for different electrode materials.

(b) *Test conclusions.*

(1) As snow flakes disappear without further treatment, after a few weeks or months, the " temporary " defects in weld metal need not cause alarm and need not normally be remedied by complicated or costly treatments.

(2) If the defect has to be eliminated within a short time after welding or quenching, the specimen can be artificially " aged " by heating to 250 deg. C. for one or two hours. This treatment eliminates snow flakes and improves ductility without modifying metal structure or other properties of the weld. Table V shows the effect of this treatment which is recommended wherever tests have to be taken shortly after welding and the appearance of snow flakes is to be expected.

(3) Numerous tensile tests on vertical welds showed rather small values of elongation and area reduction.

TABLE IV. REAPPEARANCE IN WELD METAL, AFTER AIR-QUENCH, OF SNOW FLAKES PREVIOUSLY SUPPRESSED BY HIGH-TEMPERATURE ANNEALING OR BY AGEING AT ROOM TEMPERATURE.

Annealed at 920 deg. C.		Aged at room temperature		
Treatment	Snow flake surface sq. mm.	Treatment	Snow flake surface sq. mm. A	B
Air quenched from natural state	8	Air quenched from natural state	6·5	4·3
½ hr. at 920 deg. C. slow cooling	0	60 days after first air-quench	0	1
½ hr. at 920 deg. C. then air-quenched	9	62 days, then air-quenched again	3	4·9
1 hr. at 920 deg. C. then air-quenched	7	103 days, then air-quenched again	5·9	3·5
3 hrs. at 920 deg. C. then air-quenched	12	107 days, then air-quenched again	8·5	4

TABLE V. INFLUENCE OF TREATMENT (1 HOUR AT 250 deg. C.) ON WELDS SHOWING SNOW FLAKES.

Electrode	Elastic Limit tons/sq. in.	Ultimate tensile strength tons/sq. in.	Elongation per cent	Reduction of area per cent	Remarks
D 398	27	33	29	45	A few small flakes
do. weld deposit treated	27·5	34	32	68·5	No defect
T	25·5	31·8	25	47	1 coarse flake
do. weld deposit treated	27·5	33·5	30·5	62·5	No defect
D 404	30·3	36·5	20	28	1 coarse several small flakes
do. weld deposit treated	29	37	26·5	47	No defect

TABLE VI. INFLUENCE OF HEAT TREATMENT AT 250 deg. C. ON MECHANICAL PROPERTIES AND DEFECTS OF VERTICAL-WELD DEPOSITS.

Test piece	Fracture	Elastic limit tons/sq. in.	Ultimate tensile strength tons/sq. in.	Elongation per cent	Reduction of area per cent
3·25 mm. Electrode Normal horiz. weld		23·8	29·8	29	50
Vertical mounting weld, natural state	2 flaky cracks	21·2	25·5	8	20
do. annealed 1 hr. at 250 deg. C.	no defect	24	30·2	23·2	36
4 mm. Electrode Normal horizont. weld		23·9	30·2	28	45
Vertical mounting weld, natural state	1 flaky crack	20·2	27·8	16·6	36
do. annealed 1 hr. at at 250 deg. C.	no defect	23·8	30·2	28·6	54

Fractures did not show any blow holes, inclusions, or segregations, but peculiar small plates, sometimes like snow flakes, sometimes like cracks, narrow elongated zones with dendritic texture and lighter colour than the background. The defects always occurred in parts of the welding bead which were not annealed. The analogy with rather coarse snow flakes suggested heat treatment at 250 deg. C. The treatment was successful, defects disappeared and mechanical qualities went back to normal (Table VI). This defect is therefore similar to coarse snow flakes but more dangerous as it develops in a rapidly solidified, unannealed " as cast " structure.

(4) The disappearance of snow flakes after a few minutes treatment at 400 deg. C., but only after a few hours at temperatures below 250 deg. C. also explains why snow flakes are more numerous in welds deposited at low temperatures than at high temperatures. Low-temperature welds cool down to 300 deg. C. within one minute, at a cooling rate comparable to oil-quenching and favourable for snow flake formation. If, however, a temperature of 300 deg. C. is maintained during welding, the deposit takes several minutes for cooling from 400 to 300 deg. C., and snow flake formation is much reduced if not completely suppressed.

(c) *Results from investigations.*

(1) Snow flakes in welds are not normally dangerous. They are temporary and disappear before the welded part is put into service. Moreover, snow flakes only appear under stresses near the ultimate limit, i.e. beyond the practical range.

(2) The defect can always be suppressed by a simple low-temperature treatment or by welding at sufficiently elevated temperatures.

With these results, the practical side of the problem can be considered as solved, but research continues on questions of the exact mechanism which causes snow flakes to disappear slowly at low temperatures or reappear after a second quench.

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RUSSIA

FURNACE WITH LIQUID SLAG REMOVAL

By S. M. IVANOV. (From *Ogneupori*, Vol. 11, No. 3, 1946, pp. 5-11, 3 illustrations.)

DURING 1940-1941 the Refractory Institute conducted a certain amount of work with regard to the investigation of mechanical furnaces for high rates of heat release. The first part of this research was concerned with an investigation of the movement of the coal in the firing mechanism with the help of unheated models. This investigation has been described on an earlier occasion.* The present report is concerned with tests made on a small furnace tested in the laboratory.

The design of the experimental furnace is based upon the principle of counterflow of coal and products of combustion. The fuel bed moves continually in the upward direction towards a given constant level, while the furnace gases issuing from the fuel bed pass into the lower part of the furnace chamber from whence they are discharged into the furnace proper. The distinguishing feature of this design is that the fuel bed is situated between the grate and the air stream directed against the grate. This arrangement prevents the loss of small particles of fuel and the consequent loss of combustible matter and ash from the fuel bed, which loss limits the firing rate in the ordinary type of furnace. In the arrangement under review, furnace rating is therefore limited solely by economic considerations with regard to the furnace draught.

In the ordinary type of stoker fired furnace the preparation of the fuel for combustion takes place under the influence of heat flow from the combustion zone proper, that is, in the main by heat conduction. Therefore, for a given ratio of air supply to fuel supply the combustion process becomes unstable, and in certain cases may cease altogether. In the present scheme, however, the preparation of the fresh fuel is mainly carried out by the most effective mode of heat transfer available, that is, by convection. This feature secures full stability of the combustion process at any firing rate. Owing to the positive feed employed, the fuel bed, which is effectively compressed, is kept in continuous motion. Because of this feature the possibility of holes developing in the fuel bed is automatically precluded; as is also the uneven distribution of air in the fuel bed, as any cavities that may tend to form, are continuously filled with fresh fuel.

This type of fuel feed therefore represents a combination of the features of underfeed stoker and overfeed stoker. The first experimental installation of this type, as shown in Fig. 1, was based upon the principle of ash removal in solid state. In this experimental furnace the ash and slag were periodically removed through the door 10, specially provided for this purpose; but in industrial installations ash removal will have to be carried out by a mechanical device. In order to lower the combustion temperature, water was sprayed upon the fuel bed by means of the spray nozzle ring 11. The furnace was equipped with a draught fan capable of delivering the combustion air at a maximum pressure of 6,000 mm. w.g., the fan capacity being 200 cu. m. per hour at an air temperature of 20 deg. C. The diameter of the coal conveyor screw was 100 mm. and the grate area was 0·04 sq. m., measuring 20 × 20 cm. The free flow area through the grate was 50 per cent of the grate surface.

The tests were carried out with anthracite and with gas coal of the following respective analyses.

Anthracite

Ash content	12·64 per cent
Moisture content	3·3 per cent
Volatile constituents	26·0 per cent
Lower calorific value as fired	6210 cal. per kg.

Gas coal

Ash content	15·19 per cent
Moisture content	6·0 per cent
Volatile constituents	31·3 per cent
Lower calorific value as fired	6360 cal. per kg.

In the combustion of anthracite the greatest difficulty experienced was concerned with the fuel feed. The continuous feeding of the coal towards the top was found to lead to the separation of the small fractions and to their collection in the narrow part of the retort. This trouble was further aggravated by the tendency of the anthracite to disintegrate into a fine powder at elevated temperature. The fines accumulating in the leading part of the conveyor screw were compressed into a monolithic mass which caused stoppage of the feeding mechanism. Moreover, this trouble caused the air flow resistance of the system to rise to very high values, and in some cases air flow resistances as high as 1900 mm. w.g. were recorded.

As will be seen from Table I the combustion of the anthracite took place at moderate temperatures resulting in the development of semi-gas which was passed on into the furnace proper. However, the instability of the process prevented the attainment of high firing rates, and the grate load did not exceed 400 kg. per sq. m. per hour of fuel.

*Ogneupori, No. 1, 1941.

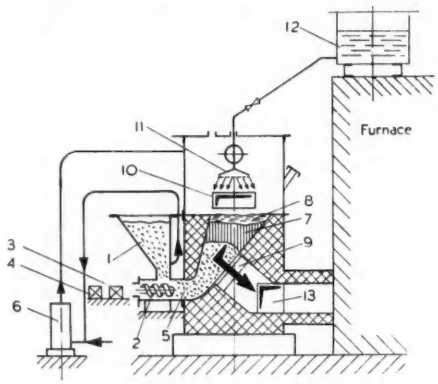

Fig. 1. **Installation with solid ash removal.**
(1) Bunker, (2) Conveyor screen, (3) Reduction gear, (4) Motor, (5) Fresh fuel, (6) Fan, (7) Combustion zone, (8) Ash layer, (9) Grate, (10) Ash door, (11) Water spray ring.

TABLE I.

Fuel	Test No. 1 Anthracite	Test No. 2 Anthracite	Test No. 3 Gas coal
Duration of test	3 h. 15 m.	2 h. — m.	4 h. 40 m.
Draught, mm. w.g.			
Maximum	1,935	700	680
Minimum	270	700	152
Average	890	700	350
Composition of combustion products, per cent			
CO_2 max.	11·0	16·8	13·6
CO_2 min.	2·0	10·9	6·0
CO_2 av.	8·0	12·0	10·9
O_2 max.	0·3	0·7	0·8
O_2 min.	0·0	0·0	0·0
O_2 av.	0·07	0·3	0·25
Average power consumption of screw conveyor, Watt	345	—	550
Average grate load, kg./sq. m., hr.	407	400	715
Gas temperature, deg. C.			
Max.	1,280	780	1,585
Min.	950	610	1,380
Average	1,000	715	1,510

TABLE II.

Fuel	Test No. 1 Gas coal	Test No. 2 Gas coal	Test No. 3 Gas coal	Test No. 4 —	Test No. 6 Gas coal + anthracite ratio 1·1
Duration of test, hrs.	4·5	3·6	8·0	5·0	7·0
Air pressure, mm. w.g.					
Max.	700	620	825	660	960
Min.	345	300	415	275	370
Average	575	515	650	547	765
Composition of products of combustion, per cent					
CO_2 max.	15·6	14·8	12·1	13·8	14·0
CO_2 min.	9·6	8·0	9·0	11·7	9·6
CO_2 av.	11·5	11·6	10·1	12·1	12·6
O_2 max.	0·8	0·4	0·5	0·3	1·6
O_2 min.	0·0	0·0	0·0	0·0	0·0
O_2 av.	0·33	0·14	0·26	0·1	0·57
Gas temperature, deg. C.					
Max.	1,660	1,400	1,450	1,400	1,300
Min.	1,460	1,300	1,220	1,200	930
Average	1,565	1,350	1,320	1,300	1,210
Average power consumption of screw conveyor, Watt	380	402	—	—	—
Average grate load, kg./sq. m., hr.	1,010	900	900	930	935

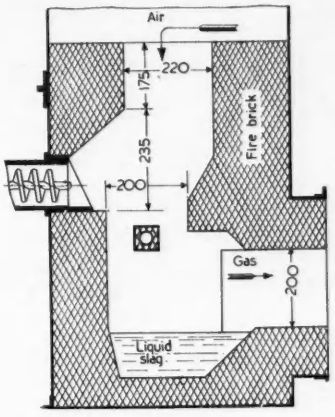

Fig. 3. **Furnace with water cooled grate bar.**

The combustion of the gas coal proved much more favourable. In this case the high temperature caused caking of the coal and the occurrence of a stable and sufficiently porous coke stratum which burned uniformly during its upward travel. Combustion was completely stable with firing rates averaging 715 kg. per sq. m. grate surface per hour.

As in the case of the anthracite, the gas coal gave off a certain amount of semi-gas containing a very small excess of oxygen. According to Table I the average temperature of the products of combustion was 1510 deg. C. and in certain cases a temperature of 1585 deg. C. was reached. Test No. 3 gave the best results both as regards capacity and stability of the process.

The test results revealed the need for a different furnace arrangement in which the slag was to be removed in liquid state. The basic design chosen for a furnace with liquid slag removal is outlined in Fig. 2. Here the principle of feeding the fuel by means of a screw conveyor is seen to be retained, but the grate is now arranged horizontally with the combustion products leaving in a downward direction. Combustion proceeds in the upper stratum and the liquid slag passes through the coke into the bottom tray which is continually heated by the hot products of combustion. The slag accumulating is periodically removed. In a first design, the grate was made of refractory materials such as several grades of fire brick and chrome-magnesite. In subsequent designs, a water cooled grate bar as shown in Fig. 3 was used, the tests being conducted with a small gas coal and with a mixture of gas coal and anthracite of the aforementioned characteristics. The firing of anthracite alone was also attempted but the same difficulties as mentioned above arose. The salient operating data of the furnace with liquid slag removal are given in Table II, while Table III gives the analysis of the products of combustion.

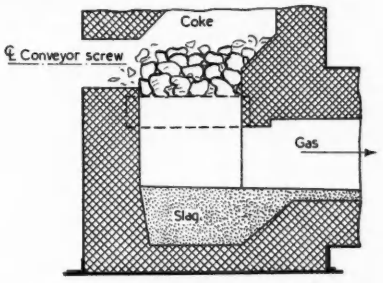

Fig. 2. **Furnace with liquid slag removal.**

In all the tests listed in Table II, operation of the furnace was fully stable and no break in the fuel bed occurred. The tests were generally broken off either because of trouble with the feed device or at the end of the working day. As will be seen from Table II, an air pressure of 600-650 mm. w.g. was required to overcome the frictional resistance of the system. The high air pressure requirements are largely due to the small grate bar clearances.

The first test was run without slag removal. Fig. 2 depicts the condition of the furnace after shutdown and cooling. The slag is seen to have overflowed into the passage connecting the combustion chamber with the furnace proper. The second test run had to be terminated because of a breakdown of the reduction gear of the conveyor screw; the furnace itself functioned properly for 3 hours 40 minutes. The slag was removed after the shutdown of the furnace. During test run No. 3 liquid slag was withdrawn three times. The first slag removal proved somewhat difficult, since the slag tray

TABLE III.

Test No.	Sample No.	Composition of combustion products, per cent CO_2	O_2	CO	H_2	N_2	Q cal./cu. m.
4	1	11·7	0·5	8·8	3·5	75·5	377
	2	13·6	0·2	10·1	3·1	73·0	400
	3	12·5	0·1	11·2	3·5	72·7	418
6	1	12·8	0·45	10·4	2·9	73·45	406
	2	13·4	0·15	10·7	2·5	72·25	403
	3	11·4	0·33	11·75	2·4	74·12	431
	4	12·6	0·48	9·8	1·9	75·22	360

TABLE IV.

Test No.	Softening Point deg. C.	Fusion Point deg. C.
3	1,210	1,370
4	1,300	1,390
5	1,130	1,410
6	1,250	1,390

was not yet sufficiently hot. Subsequent tappings were more easily effected although the slag was in highly viscous state. In part these difficulties are explained by the large loss of heat through the walls of the slag tray which was not designed for this purpose ; also, the slag showed poor fusibility. The softening point and the fusion point of the slag as ascertained in the various test runs are listed in Table IV. Actual temperatures were somewhat lower, since the slag formation took place in a strongly reducing atmosphere.

Liquid slag removal makes the construction of suitable grate bars a highly difficult and important problem. In consideration of the failures experienced with the use of refractory grate bars, water cooled bars were introduced which during operation become covered with a layer of solidified slag. A single grate bar of this type is shown in Fig. 3. This bar consists of a piece of 1 in. gas pipe. The first test run was carried out with running water, but in view of the small temperature rise of the cooling water, a second test was carried out with a circulating system. Examination of the grate bar after the test showed that the protective coating of refractory material was undamaged, and solidified slag had adhered at only a few points.

FRANCE

USE OF CONDENSERS IN MOBILE AGRICULTURAL POWER PLANTS

By E. ROQUES. (From *Électricité*, Vol. 30, No. 120, October, 1946, pp. 214-215, 4 illustrations.)

ELECTRIC motors in mobile equipment used for thrashing and other agricultural work are required to supply extra power during overload and operate at varying distances from the network transformer, so that generally they have an unfavorable power factor. The losses occurring in such cases can be considerably reduced by means of a mobile, dry-type condenser, with built-in discharging resistances.

Lower voltage drop. With the condenser the voltage drop is reduced from

$$R\,I_m \,.\, \cos\varphi + L_w I_m \,.\, \sin\varphi$$

to $$R\,I_m \,.\, \cos\varphi + L_w(I_m \,.\, \sin\varphi - I_c)$$

where R = ohmic resistance of line, per phase, L_w = apparent line inductance, per phase, I_c = current in condenser, I_m = current in motor, and $\cos\varphi$ = power factor of motor. Consequently, with the condenser the voltage will be $\Delta U = L_w I_c$ volts higher (see Fig. 1).

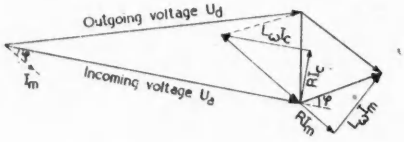

Fig. 1.

Reduced line losses. From $R\,I_m^2$ watts the line losses are reduced to

$$R\,(I_c^2 + I_m^2 - 2\,I_c I_m \,.\, \sin\varphi)$$

which is equal to a gain $\Delta W = I_c R\,(2I_m \,.\, \sin\varphi - I_c)$. This is represented graphically by a parabola, with its apex at $I_c = I_m \sin\varphi$. Thus when $\Delta W = \Delta W_{max} = R\,I_m^2 \sin^2\varphi$, the line losses reach a minimum value $R\,I_m^2 - \Delta W_{max} = R\,I_m^2 \cos^2\varphi$ (Figs. 2 and 3). There is thus a definite value for the condenser ($I_c = I_m \sin\varphi$) for which the line losses become a minimum.

Overload of transformer. The apparent power output of the transformer will change from

$$P = 3\,(U_a + RI_m \cos\varphi + L_w I_m \sin\varphi) \,.\, I_m$$

where U_a is the incoming voltage to

$$P' = 3\,[U_a + RI_m \cos\varphi + L_w\,(I_m \sin\varphi - I_c)] \,.\, \sqrt{I_c^2 + I_m^2 - 2I_c I_m \sin\varphi}$$

when the condenser is used, so that for $I_c = I_m \sin\varphi$ the total transformer output will be reduced by

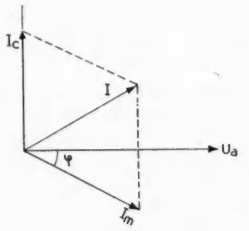

Fig. 2.

$$\Delta P = 3\,I_m \,.\, [(L_w I_m \sin\varphi + (1 - \cos\varphi)\,(U_a + RI_m \cos\varphi)].$$

Numerical example. An overhead line, three-wire system 50/10 (length 700 meters, distance between conductors 80 cm.) feeds a 25 H.P. motor of modern design, η = 80 per cent, $\cos\varphi$ = 0·75, working voltage 380 volts, line resistance 0·62 ohms, apparent inductance 0·27 ohms, I_m = 37·4 amperes. Application of the foregoing formulæ gives the following results :

	Network voltage U_a	Line current intensity	Line losses watts	Line losses per cent	Apparent power output of transformer
Without condenser	434 V.	46·5 A.	4,050	17·6	34·8 kVA.
With condenser	417 V.	35·0 A.	2,270	9·8	25·2 kVA.

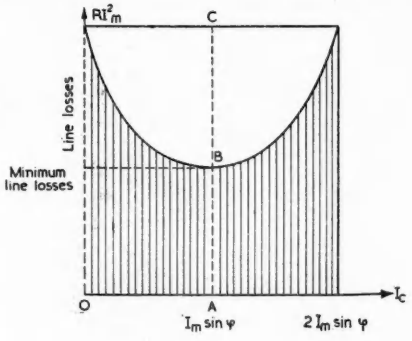

Fig. 3.